POWER PLANT ENGINEERING

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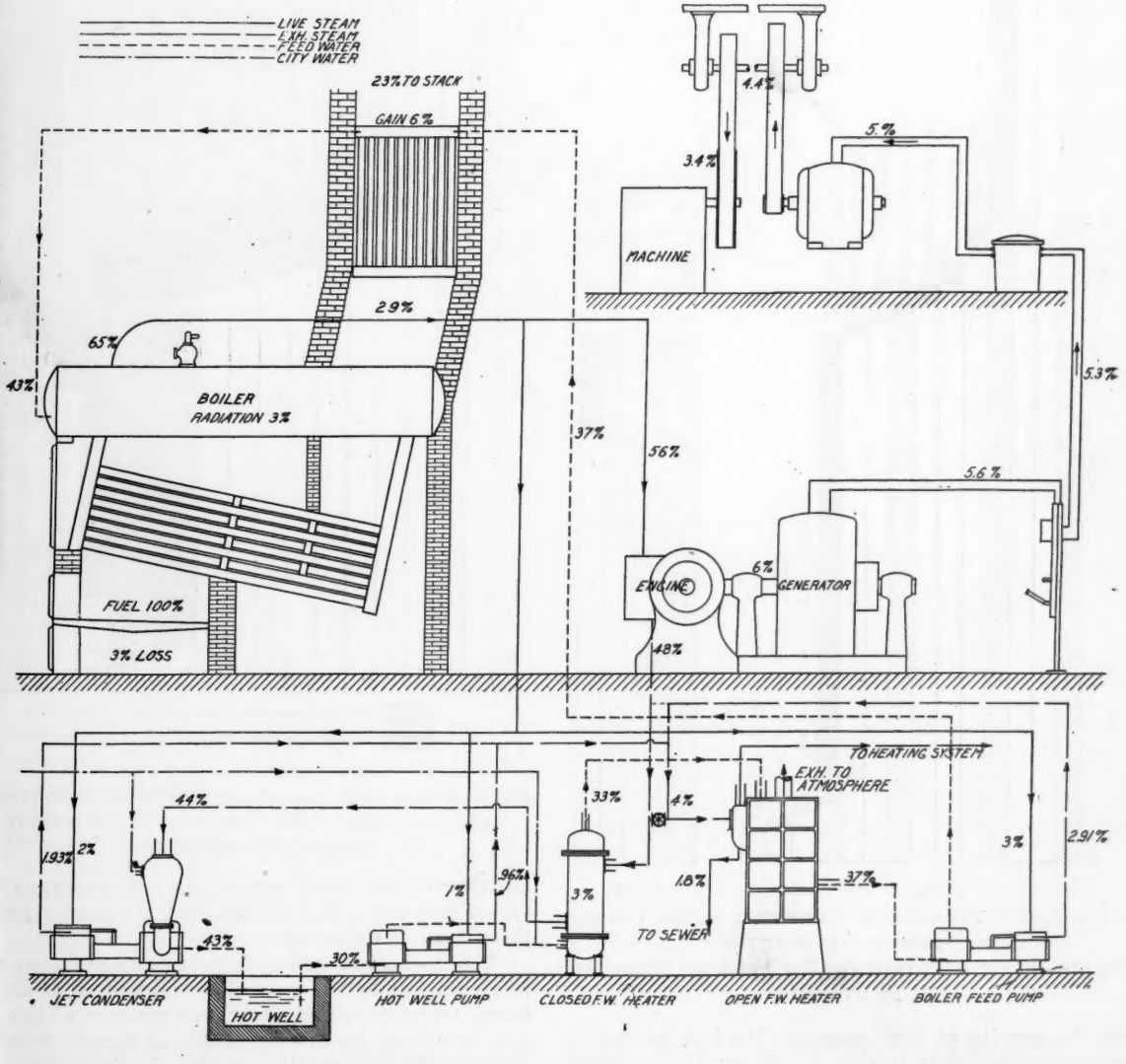
Energy Distribution Diagram from Fuel in a Well-Equipped and Operated Steam Plant



Chasing the Wastes

WHERE DOES the heat go? It comes to us in the fuel and costs good hard dollars. We buy it and use it to get certain products, cloth, guns, shoes, sugar, comfortable housing. All these demand fuel; all are needed by our soldiers over there and our civilians over here—and in ever increasing quantities—even greater than we can supply.

Waste of fuel or heat or its other self energy, either mechanical or electrical, is waste of food, clothing and housing. It is unpatriotic. It is inhuman. For there will be scarcity of these essentials the best we can do, and anything below that best will result in privation and hardship for those who cannot get these essentials because of the waste.

And so it becomes vital, not only as a matter of economy, but for common humanity's sake, that no fuel be wasted, no heat unit be allowed to stray, that can possibly be guided into productive work.

To give an idea of what becomes of the heat in fuel, the diagram on p. 1 has been drawn. Starting with 100 per cent of heat in the fuel as fired to the boiler, there are boiler losses to ashpit by unburned fuel, to the stack by hot gases, and by radiation from the heads and setting. The balance passes in the steam to the main, where some more is lost by radiation, some diverted to pumps, and the remainder conducted to the engine. There some losses are found in radiation and friction, an unfortunately small part of the heat energy is converted into mechanical energy, and the greater part is thrown out of the exhaust.

The mechanical energy is most commonly converted into electrical by the generator, transmitted to switchboard, transformer and motor, and reconverted into mechanical energy for driving machinery through belts and shafting, the final result in a typical case as shown be-

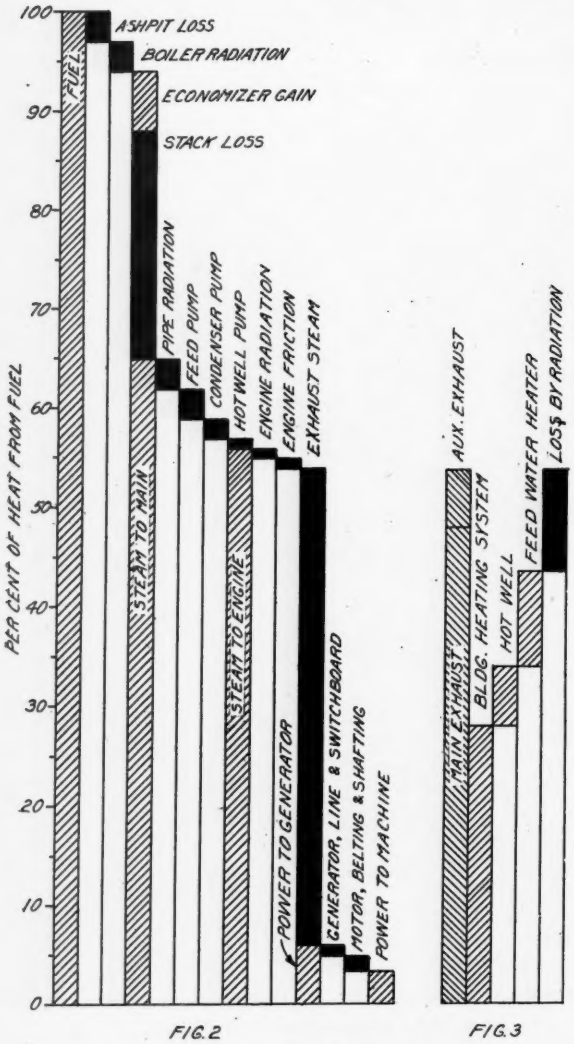


FIG. 2. TYPICAL THERMAL EFFICIENCY CHART, LOSSES INDICATED IN BLACK

FIG. 3. CHART ILLUSTRATING HOW EXHAUST STEAM MAY BE UTILIZED

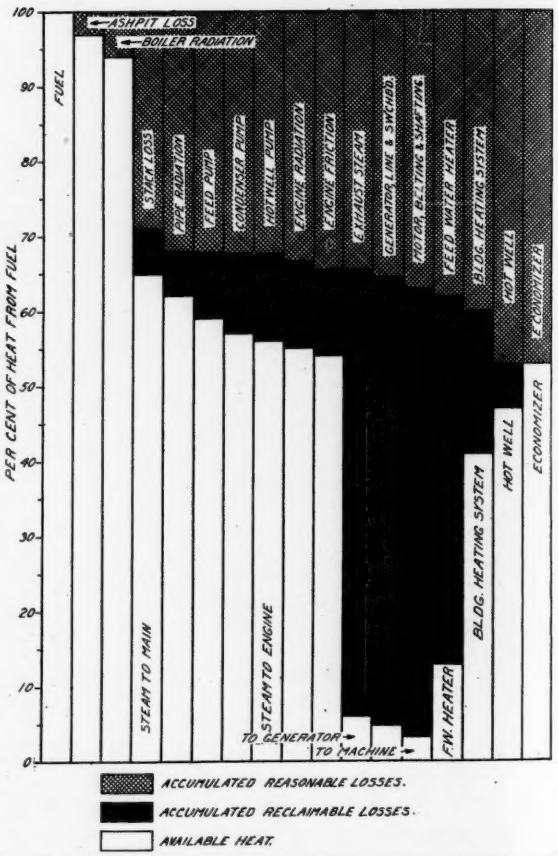


FIG. 4. WHERE DOES YOUR PLANT STOP RECLAIMING LOSSES? THE BLACK AREA ABOVE INDICATED STOPPING POINT SHOWS UNNECESSARY LOSS

ing that of 100 B.t.u. in the furnace, 3.4 B.t.u. are utilized at the driven machine.

While the amount of energy that may be saved by reducing losses in generator, electric lines, transformer, motor and shafting may seem small, it must be remembered that a heat unit saved there represents a saving of 16 B.t.u. in the boiler furnace.

Going back to see where some of our losses may be recovered, at the stack, an economizer may catch a goodly amount of heat, and transfer it to the feed water, thus saving fuel; the exhaust steam from engine and pumps may be passed through heaters to heat the feed water, and the balance of the steam used for heating buildings or in manufacturing processes. Condensed steam may be passed to the hot well, cleansed of oil and used as feed water, thus saving a considerable part of the heat which it carries.

To make sure that we are doing our best to save fuel, every step, machine and piece of equipment must be gone over, its condition ascertained and its performance checked up to make sure that losses are at a minimum, efficiencies and reclaiming methods at a maximum.

Diagram Fig. 2 shows the proportion in which heat is lost at various points and Fig. 3 the proportion in which it is reclaimed.

In the following pages, each step of the process is taken up in detail to show how wastes may be hunted out and stopped. On with the chase:

PROPORTION OF POWER PLANT LOSSES

Hand-fired boiler H. R. T.—	Per Cent
Fuel in Furnace	100
Ashpit loss	3
Radiation loss	3
Gases to economizer	29
Steam to main	65
Economizer—	
Gain to F. W.	6
Gases to stack	23
Steam Main—	
Piping radiation loss	3
Branch to boiler feed pump	3
Branch to condenser pump	2
Branch to hot well pump	1
Main to engine	56
Engine—	
Radiation loss	1
Friction loss	1
Exhaust steam	48
Power to generator	6
Generator—	
Resistance and friction loss	0.3
Line to switchboard loss	0.1
Power to switchboard loss	5.6
Switchboard—	
Resistance loss	0.2
Line to transformer loss	0.1
Power to transformer loss	5.3
Transformer—	
Resistance and iron loss	0.2
Line to motor loss	0.1
Power to motor loss	5
Motor—	
Resistance and friction loss	0.5
Belting to shafting loss	0.1
Power to shafting loss	4.4
Shafting—	
Friction loss	0.9
Belting to machine loss	0.1
Power to driven machine	3.4
Boiler Feed Pump—	
Ex. steam loss	2.91
Condenser pump loss	1.93
Hot well pump loss	0.96
Atmospheric Ex. Piping—	
Delivered to F. W. heater	4
Oily drips to sewer	1.8
L. P. Ex. Piping—	
Radiation loss	1
L. P. heater	3
To condenser	44
Radiation loss	1
To hot well	43
Hot Well—	
Radiation loss	13
To feed water	30
L. P. Heater, Closed—	
Gain to feed water	3
Atmospheric Heater, Open—	
Gain to feed water	4
F. W. return to boiler, condensing	43
Alternative Noncondensing and Heating—	
Atmospheric ex. line takes	53.8
Radiation loss	1.2
Heat to F. W. heater	52.6
Atmos. F. W. Heater, Open—	
Radiation	1
To F. W.	9.6
To heating system	43
Heating System—	
Radiation	2
Useful heating	28
Hot well	12
Hot Well—	
Radiation	6
Feed water	6
Feed Water—	
From hot well	6
From heater	9.6
From economizer	6
Returned to boiler	21.6

TEN YEARS ago, only, Wilbur Wright succeeded in flying 30 mi. in 40 min. Now we have planes capable of carrying a dozen or more men. One, used since 1916, recently made a flight of 500 mi.; and another, after having the lower plane stripped of canvas by a high explosive shell, flew 60 mi. to its own lines. It is difficult to foresee what will be accomplished with another 10 yr. of development along lines useful to peace activity.

Testing of Coal

SIMPLE TESTS TO BE EASILY MADE WITH INEXPENSIVE APPARATUS TO DETERMINE THE QUALITY OF COAL SUPPLIED TO A PLANT. BY HENRY KREISINGER

THIS ARTICLE deals with methods of testing coal, to determine the quality of a shipment consisting of one to several carloads.

Testing work consists mainly of two parts, namely, collecting and reducing the sample, and the testing of the reduced sample. Of the two parts, the collection and reduction of the sample requires the greater care and judgment, particularly if the coal contains a large percentage of lumps. This fact is not as generally appreciated as it should be, and as a result much time is wasted in testing worthless samples.

In order for a test to be of any practical value, the sample must be collected so that it is representative in every way of the quality of the entire shipment. If the collection of the sample is made carelessly and the sample does not represent the entire shipment, the testing of the sample is not only useless, but may lead to unjust decision.

TIME OF SAMPLING

USUALLY the best time to collect the sample is when the shipment is being unloaded. If the coal is unloaded over a conveyor and is run through a crusher, it is better to collect the sample at some point after it has left the crusher. The coal is then of more uniform size and it will be easier to obtain a representative sample. If laboratory determinations are desired of a coal burned on a boiler test, it is best to collect the sample from each cartful, or every other cartful, as the coal is being delivered over the scale to the firing floor.

SIZE OF SAMPLE

THE SIZE of the sample that should be collected varies with the size of the coal. When sampling run-of-mine containing a large percentage of lump, the sample should weigh at least 1000 lb. When sampling screenings a 500-lb. sample may be sufficient.

Usually an ordinary shovel or coal scoop is the most convenient tool for collecting the sample. When sampling very lumpy coal, the larger the shovel the better. Small shovels, such as are used for feeding coal into a stove, are not recommended.

METHOD OF COLLECTING SAMPLE

WITH ALL sizes of coal, it is preferable to collect the sample by complete shovelfuls, because by such large increments it is easier to get the proper proportion of lump and fines than if small increments are taken with the corners of the coal scoop, or if a small hand-shovel is used for sampling. Dipping a corner of a coal scoop at several places in the coal pile or a cart to obtain one shovelful is sure to result in the collection of more fines and less lump than contained in the average of the coal, and cannot be considered a representative sample.

Most of the high grade eastern coals are soft and crumble easily. The lumps are held together by thin layers of slate or bone coal which does not crumble easily; consequently, the lumps will contain more ash than the fines. On the other hand, coals like those from the Illinois field are hard and stay in lumps without the binding strength of the slate. In fact, most of the impurities are softer than the coal and crumble easier than the coal. Consequently, the fines of these coals contain higher percentage of ash than do the lumps. Furthermore, the fines of all bituminous coals hold more moisture than the lumps when exposed to rain. For these reasons care must be exercised in collecting the sample that the same proportion of lumps and fines are collected as contained in the coal.

A sample should not be collected from the surface of coal piles, bins, cars, or barges. Invariably the surface contains more lumps than the interior of the pile. It is advisable to dig 3 or 4 ft. into the bulk of the coal when obtaining a sample. Such digging should be done and the sample collected at several places in the pile.

It is difficult to give detailed instructions for sampling coal that will apply in all cases. When sampling coal, one must bear in mind that the sample must be representative of the entire shipment and judgment must be exercised to make it so.

REDUCING THE SAMPLE

AFTER the sample has been properly collected, the coal must be crushed and the sample reduced to a convenient size for laboratory manipulation. This crushing and reducing should be done on some hard, smooth surface sufficiently large to hold all the coal. A good smooth concrete floor can be used for this purpose, but if this is not available, a ⅛ to ¼-in. steel plate should be obtained. The sample is successively crushed and reduced to one-half by the quartering method or by discarding alternate shovelfuls. With large samples it will be found more convenient to reduce the sample to about 500 lb. by discarding alternate shovelfuls, and then finish by quartering.

After the lumps have been crushed to the required size, the coal is shoveled into a conical pile and well mixed by taking the coal all around at the base of the pile and depositing it at the top. The coal is then taken, a shovelful at a time, and spread in a ribbon about the width of the shovel and 5 to 10 ft. long. The spreading of each shovelful starts alternately at the two ends of the ribbon which gradually grows into a pile. When all the coal has been placed in this long pile, the sampler, starting at one end of the pile and at the bottom of it, takes shovelfuls and deposits them alternately on two separate piles, advancing in the same direction around the long pile until all the coal has been removed to the two piles. One of the two piles is then rejected and the coal in the other is crushed, mixed, and again made into a long pile and reduced by shoveling it into two piles.

To reduce a sample by quartering, the coal is shoveled into a conical pile and well mixed by taking the coal at the base of the cone and placing it at the top. The cone is then flattened down with a shovel and the circular

pile is marked by two crossing lines into quarters. Two opposite quarters are rejected and the space vacated is swept clean; no fines should be left behind. The remaining two quarters are crushed and mixed and again reduced by quartering, and so on until only 5 lb. of 3/16-in. coal is left. This coal is then placed into a jar or can, sealed and transmitted to the laboratory. The crushing and reducing of sample should be done in a place where the coal would not be exposed to sun or to rain. The work should be done as quickly as possible so that there would be as little change of moisture as possible.

The following table shows the successive sizes of coal and the weight of sample required by the U. S. Bureau of Mines in the process of reduction of the gross sample.

Weight of Sample in Pounds	Largest Size of Pieces Allowable in Sample Before Quartering
Gross Sample 1000 or more	1 inch
Gross Sample 500	¾ inch
Gross Sample 250	½ inch
Gross Sample 125	⅜ inch
Gross Sample 60	¼ inch
Gross Sample 30	3/16 inch

Preparation of Sample in Laboratory

When the 5-lb. sample reaches the laboratory, the coal is spread to a depth of 1 in. on a metal sheet pan and weighed. It is then placed in an air drying oven through which passes air at 30-35 deg. C. (85-95 deg. F.). The coal is kept in this oven until the change in weight between two weighings, 6 to 12 hr. apart, is less than ½ per cent. The reduction in weight is called the "air drying loss." It is always less than what is known as the moisture in coal. It has no significance so far as the quality of coal is concerned; the sample is air dried in order that the coal may not lose moisture while it is handled in the laboratory.

The air dried sample is then crushed in a roll crusher to a 10-mesh size, mixed, and reduced by the quartering method to 500 grams. This reduced sample is ground in an Abbe ball mill to pass through a 60-mesh sieve. The Abbe ball mill consists mainly of an earthen jar of about 1 gal. capacity and is partly filled with small pebbles. The jar is sealed air-tight and rotated by machinery at about 60 r.p.m. and the coal is ground by the action of the pebbles. Bituminous coals require about ½ hr. and anthracite about 2 hr. Large particles that still remain in the coal after this period should be crushed to the required size by hand. The sample is then reduced by successive quartering to 50 grams and placed in a 4-oz. bottle with a rubber stopper. The hand-crushing and reducing of this sample should be done as quickly as possible to prevent absorption of moisture from the atmosphere; a skilled man can do it in less than 3 min. The coal in the bottle is the coal used in making chemical analysis, but, before taking any coal out of this bottle for chemical determinations, the contents should be thoroughly shaken to insure uniformity of the coal.

Proximate Analysis

Proximate analysis consists of the determination of moisture, volatile matter, ash, and the fixed carbon. Of these, the moisture and the ash are the most important and most useful in testing the quality of coal.

The apparatus needed for making proximate analysis consists of the following: Chemical balance sensitive at least to 1/1000 gram., drying oven, desiccator, 0.6 cu. in. platinum crucible with a platinum cover, porcelain capsule 1⅜ in. diameter and ⅞ in. deep, Meker burner No. 3, muffle furnace.

All the above apparatus are stock articles on the market.

Moisture

For the determination of moisture, 1 gram of the powdered air-dried coal is weighed out in the porcelain capsule and placed for 1 hr. in the drying oven, which is kept at 105 deg. C. (221 deg. F.) by a boiling mixture of two parts of glycerine and one part of water. Air previously dried by passing through concentrated sulphuric acid is passed over the coal at such a rate as to change the air in the oven two to three times a minute.

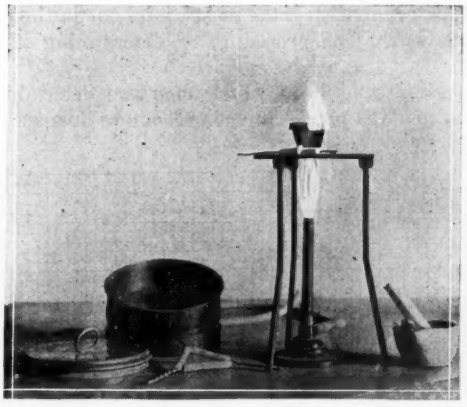


FIG. 1. DRIVING OFF VOLATILE COMBUSTIBLE OVER A GAS FLAME

After 1 hr. in the oven, the capsule is cooled in the desiccator and weighed. The loss in weight is called the moisture at 105 deg. C.

The total moisture in the coal as received consists of the sum of the moisture in the air dried coal plus the moisture at 105 deg. C., both reduced to the same basis.

Volatile Matter

Determination of the volatile matter is entirely a conventional method. The coal does not contain volatile matter in the same sense as it does moisture. Under a heat treatment the coal substance breaks down, and part of it escapes in the form of gas and tar. This gas and tar does not exist as such in the original coal. The amount of the gas and tar escaping from the heated coal depends not only on the character of the coal, but also on the rate of heating and the temperature to which the coal is heated. Therefore, to have true comparison between different coals as to the volatile matter, the coals must be subjected to the same heat treatment. The Bureau of Mines heats one gram of coal 7 min. in an electric furnace which is kept at 950 deg. C. (1742 deg.

F.), which method of heating gives most uniform and consistent results. Fairly consistent results can be obtained by heating the coal in the flame of a No. 3 Meker burner, the flame being adjusted from 6 to 7 in. in length. This method of heating is more readily available in the average coal laboratory. The procedure of determining the volatile matter is as follows:

One gram of the powdered coal from the 4-oz. bottle is placed in the platinum crucible which is closed with a well-fitting capsule cover. The crucible is then placed in the adjusted flame of the Meker burner in such position that the bottom of the crucible is ¾ in. above the top of the burner, and a sheet iron chimney is placed around the burner to prevent draft. After 7 min., the crucible is removed from the flame, cooled and weighed. The loss in weight is the volatile matter plus moisture in the air dried coal. The operator may make several such determinations to satisfy himself of the consistency of his results.

Ash Determinations

Ash determination is the most useful one in the proximate analysis. The procedure of determining it is as follows:

The same sample on which moisture determination has been made is placed in the cool muffle furnace. The

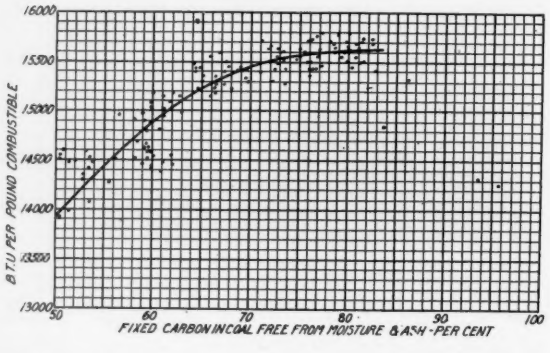


FIG. 2. HEAT VALUE CURVE FROM PROXIMATE ANALYSIS

temperature is then gradually raised to about 750 deg. C. (1382 deg. F.). The content of the capsule is occasionally stirred until all the carbon particles have disappeared. The capsule is cooled in the desiccator, weighed, and placed back into the furnace. This burning and weighing is repeated until two successive weighings give the same results. The residue in the capsule may be considered for all practical uses as the ash in the coal.

Fixed Carbon

Fixed carbon is the difference between the weight of the residue left after the determination of the volatile matter and the ash.

Determination of Heat Value from Proximate Analysis

It has been proposed to reduce the heat value of a coal from the proximate analysis by means of a simple curve. The best results can be obtained from the percentage of fixed carbon in the combustible. The combustible is the coal freed from moisture and ash.

Figure 2 shows such a curve obtained by plotting about 150 coal analyses with the fixed carbon as abscissas and the heat value as ordinates. It is apparent, however, that such curves can be used only with a limited range of 65 to 85 per cent of fixed carbon. Within this range the heat value may be determined by this method with a possible error of 1.5 per cent. Outside of this range an error of 4 to 5 per cent is possible. Inasmuch as a large part of the steaming coals used in this country have the percentage of volatile matter lower than 65, such curve can have only a limited usefulness. Much better results can be obtained from the published analyses of American coals.

Obtaining Heat Value of Coal When Its Origin is Known

Coals coming from the same seam and the same locality have nearly the same heat value per pound of combustible. Therefore, if the origin of the coal is known, its heat value can be obtained from analyses on similar coals. All that is then necessary is the determination of the percentage of moisture and ash of the coal sample, and to find the heat value for such moisture- and ash-free coal in the tables of analyses. The U. S. Bureau of Mines has published thousands of reliable analyses of coal from all parts of the country. These analyses, together with the information of the origin of coal, are conveniently tabulated in bulletins Nos. 22, 23, 85 and 123 of the Bureau of Mines, which supply the cheapest, simplest, and perhaps the most reliable means of determining the heat value of a coal. They can be obtained either free of charge from the Bureau of Mines, or, in case the edition for free distribution is exhausted, through the Superintendent of Documents, Government Printing Office, Washington, D. C., at a small cost. Everybody interested in the composition of coal should possess a set of these bulletins.

It should be remembered that only the heat value per pound of moisture- and ash-free coal is nearly constant; the heat value of the coal as received will be affected by the percentage of moisture and ash, which factors may vary over a wide range with the same coal. To obtain the heat value of the coal "as received" from the known value of the "moisture- and ash-free" coal, it is necessary to multiply the latter by the percentage of the moisture- and ash-free coal and divide by 100.

When the heat value of coal as fired is known, the probable evaporation can be computed, if the approximate efficiency of the boiler plant is also known. Surely these days the engineer or the manager of the plant should know what is being done in the boiler-room. In the average plant, 65 to 70 per cent of the heat in the coal is utilized in making steam, the remaining 30 to 35 per cent is lost in various ways. Therefore to obtain the evaporation, the heat value of the coal as received is multiplied by 0.65 to 0.70, as the case may be, and the result is the quantity of heat that the boiler absorbs from each pound of coal. By dividing this quantity of heat by 970, the equivalent evaporation per pound of coal as fired is obtained. If one wishes to know the

actual evaporation he simply divides this equivalent evaporation by the factor of evaporation.

The efficiency that can be obtained with a given coal in a given plant depends somewhat on the size of the coal, the percentage of ash and the percentage of moisture. When the coal contains a large amount of slack, the flow of air through the fuel bed is hindered. The surface of the fuel bed may fuse and form a hard crust which may have to be broken by a fire tool. To break this crust requires that the fire door or poking door be opened, which allows large excess of air in the furnace. In addition to this, the breaking of the fire may be the cause of excessive clinker, which must be removed by more frequent cleaning of fires and thus further be a cause of large excess of air and the resultant heat losses.

Large percentage of ash in coal interferes with the proper air supply, and makes frequent cleaning of fires necessary, hence, the heat losses caused by high ash are largely due to large excess of air. There is, also, larger loss of the combustible in the refuse. Because of quicker accumulation of ash on the grate and the consequent reduced flow of air through the fuel bed, the rate of combustion that can be obtained with a given draft is lower. While the fires are being cleaned the boiler's steam production is almost suspended, so that the more frequent the cleanings, the oftener the boiler is kept out of operation. Therefore high percentage of ash in coal causes a low capacity of boiler.

Moisture in coal reduces the efficiency of a boiler to a small extent because heat is required to evaporate the water in the coal. The loss due to this cause amounts to 1 to 2 per cent for every 10 per cent moisture in coal. Excessive moisture may have a detrimental effect on the combustion of the coal, but high moisture is not nearly so objectionable as high ash, because the moisture does not accumulate in the furnace and no special effort is needed to remove it.

Fusing Point of Ash

Within reasonable limits, the quantity of ash is not nearly so objectionable as the clinker-forming property of the ash. The clinker is formed by the melting of the ash. Prevention of the clinker consists of keeping the temperature in the fuel bed below the melting point of the ash, or, by purchasing coals having ash with melting points above the temperature of the fuel bed. A large number of the coals mined in the United States have ash with a comparatively low melting point—below 2400 deg. F. It would be rather a short sighted policy to attempt to exclude such coals when buying fuel for steaming purposes, because the coals with ash having high melting point could not supply the demand, and in many localities such coals would not be available. Besides the tests of the melting point of ash are rather difficult to make, and the methods have not as yet been standardized, to make such tests reliable. Wherever the market is supplied with coal having ash with low fusion point, the furnace equipment should be so chosen and so operated that the fuel temperature remains low enough to prevent troublesome clinker. This doesn't mean that the furnace must be operated with high excess of air. It is a known fact that in a hand-fired furnace troublesome clinker can be prevented by running a light fire, and at the same time the lighter fire gives higher percentage of CO_2.

The Effect of Different Elements on the Formation of Clinker

The effect of different elements in the ash on the formation of clinker is not definitely known. It can be said that, in a general way, a large percentage of lime, iron, sodium, and potassium compounds increases the fusibility of ash and the clinker trouble. But, even if the effect of these compounds were definitely known, such knowledge could not be used readily because ash analysis is rather difficult and expensive to make.

High sulphur is frequently blamed for clinker trouble, but it is doubtful whether it has anything to do with clinker. The sulphur usually occurs in combination with iron, and it is very likely that it is the latter which is responsible for the clinker. Many coals with low percentage of sulphur are noted for their clinker trouble.

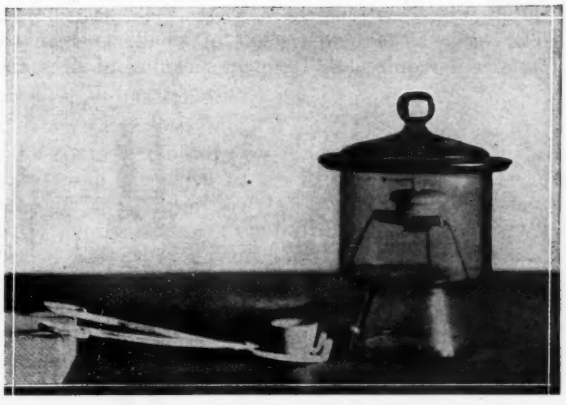


FIG. 3. DESICCATOR USED TO KEEP SAMPLE DRY WHILE COOLING BEFORE WEIGHING

It seems that the distribution of ash in the coal matter has even greater effect on the clinkering property than the chemical composition of the ash. Ash which is finely distributed through the coal is much more liable to give trouble from clinkers than ash occurring in definite layers that are easily distinguished from the coal. It would seem that such larger pieces of ash have a tendency to sink to the grate and thus separate themselves from the incandescent coal and therefore are not exposed to high temperatures to such extent as are the small pieces of ash finely distributed through the mass of burning coal.

Hand Picking

When the supply of coal comes continuously from the same seam or same mine, a good idea of the quality as to ash of different shipments can be obtained by picking the visible impurities, such as slate and sandstone, by hand and weighing them. Several wheelbarrowfuls representing the carload can be spread over the boiler-room floor or some other clean place and the visible im-

purities picked out. If only a small place is available for picking, the coal can be spread one wheelbarrowful at a time. The coal as well as the impurities should be weighed, and the percentage of the impurities determined. Of course this method of determining the impurities is applicable only to run-of-mine and coals containing a comparatively large percentage of lumps. It cannot be applied to coal containing much slack or fine screenings. Although this method gives only a relative measure of the quality of coal, it furnishes valuable information to the engineer of the quality of coal shipments, and should be used whenever possible, especially by the small user. It may save many misunderstandings between the coal shipper and the user.

Effect of Washing Coal

Washing removes from the coal only the impurities that are loose or can be loosened by running the coal through a crusher. Usually all coal is run through a crusher before washing. The finer the coal is crushed, the better the separation of the impurities from the coal and the larger the percentage of reduction of ash by washing.

The washed coal, on account of smaller percentage of ash, will require less frequent cleaning of fires and

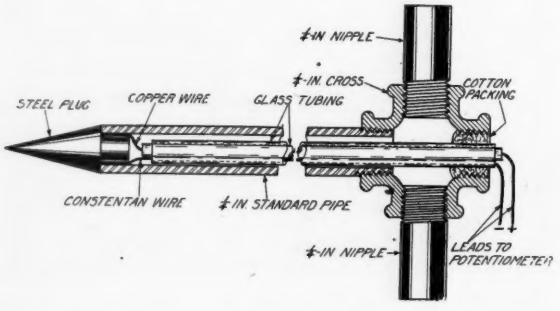


FIG. 4. THERMO COUPLE FOR MEASURING TEMPERATURES IN A COAL PILE. THE CROSS AND NIPPLES FORM A HANDLE TO FORCE THE COUPLE INTO THE PILE

makes possible higher boiler capacity, but the efficiency of the boiler is not as a rule appreciably increased. In hand-fired furnaces the washed coal may cause more clinker trouble than unwashed coal. This is probably due to the fact that only the large pieces of impurities are removed. The ash which is uniformly distributed through the coal remains in it, and, on account of its distribution, stays with the coal in the hot fire and is therefore more apt to melt.

Effect of Weathering on Coal

When stored in air, coal deteriorates by oxidation, losing from 1 to 3 per cent of its heat value. The higher grades of coal lose less than the lower grades. When coal is stored under water, the loss is negligible. By far the largest loss occurs during the first four weeks after the coal has been taken out of the mine. These statements are true only if the coal does not heat in storage. If it heats, the loss increases rapidly, and if allowed to go on will lose all of its volatile matter. Under normal conditions the deterioration of coal in storage is so small that it is not considered a serious drawback to storage of coal. The danger of the coal heating up and burning is the point to be carefully guarded.

In some cases the weathering of the coal has a beneficial effect on the behavior of the ash in the fire. Some coals which gave considerable trouble from clinker when burned soon after they were taken out of the mine, gave practically no trouble after being stored in open air for three months.

Precautions in Storing Coal

It is safer to store coal that is comparatively free from slack. If coal contains too much slack, it is advisable to screen off the slack, store the screened coal, and burn the slack or store it separately. This is probably the most important precaution to avoid fire.

Coal should be stored in as small piles as the available space will permit. Small piles are not so apt to start a fire as large ones, and if the fire is started it is more easily extinguished in the small pile. However, small piles are not an absolute guarantee against fire.

When storing coal in a pile, driving a wagon over the coal pile should be avoided. Heating is likely to start where the coal is pressed down by the wheels of the wagon. If there is no other way to get the coal on top of the pile, a trestle work should be built for the wagon or cart.

All external sources of heating coal, such as steam pipes, furnace walls, or direct heat from the sun should be avoided. If possible, coal should be stored in cool weather. It is not advisable to build up the coal storage pile in layers and allow the surface of each layer to become heated a day or two by a hot summer sun, and then cover it with another layer. This heating by the sun may store enough heat in the coal to give trouble early in the fall. Last summer a considerable quantity of coal was stored during the steady hot August weather and there seem now to be more coal pile fires than usual. When the coal once starts burning, the best way to stop it is to re-handle the coal, giving it a chance to cool as it is shoveled into another pile. Burning coal should be used immediately if at all possible. Pouring water on a burning coal pile may check the fire for a short time but seldom extinguishes it.

If there is a suspicion that the coal pile is heating, the temperature inside the coal pile should be determined with a thermocouple placed in a ¼-in. iron pipe, driving the pipe to various depths in the places of the pile where spontaneous combustion is suspected. In this way the hot coal can be located and removed in time. As long as the temperature stays below 150 deg. F. there is no immediate danger from fire. A thermocouple made of copper and constantan wire will answer the purpose very well. The wire is inexpensive and couples made of it will give the temperature with sufficient accuracy for all practical purposes. Figure 4 shows a convenient form of such a thermocouple. A couple 12 ft. in length will be found sufficient for most cases. A suggestion on how to calibrate and use such couples is given in Bureau of Mines Bulletin No. 145.

Boiler Room Inspection

OBJECTS OF INSPECTION. POINTS TO BE GIVEN ATTENTION. INDICATIONS OF WASTE DURING OPERATION. BY ROBERT H. KUSS

FIRST OF ALL, a dirty boiler room offers no encouragement or prospect of cleaning up leaks or correcting operating faults; indeed, dirt is not only the evidence of but inefficiency itself. Let the inspection service clean up "things out of place" and an excellent start has been made for better things.

There are some inspections that simply require looking at the thing or performance investigated in order to give an idea of its effectiveness in the direction intended, while others are not open to such simple and ready means. For the latter classes it is inevitable that contrivances in the way of measuring, weighing or gaging must be available, and engineers who are required to work without at least one differential draft gage, a flue thermometer, ordinary platform scales and some form of water weighing or measuring device are indeed badly off.

There are two conditions of operation which need to be considered separately, namely, "shut-down" and "in service" inspections, because what may be possible of investigation under one may not be so under the other. However, every inspection revealing defects during a shut-down investigation will, except for safety features, be indicated in one or more ways during operation, and vice versa. In general, shut-down inspections afford the only means for gaging the safety features while service inspections reveal the effectiveness of design and installation faults. Also, a service inspection may show that a fault exists somewhere, while it requires a shut-down inspection to locate exactly where the fault lies.

Inspections of whatever kind have for their purposes the investigation, and subsequent correction of faults so revealed, of distinct features of boiler room conditions, namely:

I. Safety, or the certainty of the apparatus to function without mishap.

II. Economy, or the ability to deliver and probability of the apparatus and its operatives delivering adequate steam return for the fuel, material and labor expended.

III. Design improvement, or the revealment of defective proportions which can be changed with the certainty that available economy of operation is favorably affected, thus avoiding costly experimentation.

In explanation, it may be stated that the second division, "Economy," is inseparably associated with "Capacity" in its general aspects, though it does not follow that every move toward greater economy will have a corresponding reflection in greater capacity.

SAFETY

IT IS customary, and not without reason, to assign the judgment for the safety of boilers to governmental authorities and private insurance companies. However, the operating engineer must feel he has a large responsibility in these matters even with the most favorable outside assistance, as can be readily understood when it is realized that such outside inspections are made annually whereas serious trouble may develop in a few days.

The most prolific sources of boiler failures, outside of gradual deterioration of metal, have to do with:

(a) Dense scale formation on tubes, especially in the hottest zones.

(b) Oily deposits on heating surfaces, most dangerous in hottest zones.

(c) Failure to obtain adequate feed water.

(d) Failure of supporting structures.

(e) Failure of pipe line equipment to operate properly.

It is significant that none of these five items constitutes the main concern of outside inspection service. Remember that the boiler unit is prepared for such inspection by overcoming the very defects listed, some of them not discernible except during operation, and it will be seen that the best value of outside inspection comes from the preparation rather than the inspection itself, barring, of course, the matter of corrosion internally and externally.

Item (a) is manifestly a matter of inspection by the chief engineer himself (not to be delegated to a subordinate) who must see to it that all tubes are cleaned thoroughly. The presence of spots of scale may be the very thing that causes the burning and subsequent failure of tubes. The importance of circulating tubes or nipples outweighs that of the heating surface tubes when it comes to the matter of safety. Where bent tubes are met with, the obligations of the inspector to guard against scale are particularly important.

When it is not necessary from the nature of the water to turbine tubes throughout at each shut-down inspection should still be made, by removing a typical vertical row of handhole plates of a horizontal water tube boiler, selecting a different row each shut-down. The fact that scale has been removed from its place of formation is not sufficient; it must be taken out of the boiler structure entirely, from the mud-drum of water-tube boilers, bottoms of headers, etc.

Oily deposits (item b) are the most insidious of trouble makers. When traces of oil are found (shown usually most markedly at the water line) the important thing is to find its source. Spare no effort in eradicating the oil at once. When a boiler, usually well behaved, develops an erratic or "jumpy" water level in the gage glass, it is time to search for oil; there is then more than an even chance that oil has found its way into the boiler.

Boiler feed piping (item c) should be inspected at every shut-down. It pays to use brass interior feed piping. Exterior feed and water column piping ought to be connected up with crosses and tees permitting thorough cleaning without disconnecting.

Boiler supports (item d) require constant watching if subjected to heat. If the boiler rests on brick work

(as on lugs) the shut-down period is the only time when a good job of inspection can be done; don't run any chances with bulging brickwork. If supported on columns running alongside of the furnace, particularly within the inside walls of a battery setting, provide air circulation facilities along the column. This means places for both ingress and egress for air and a passageway between; keep these air openings and passageways clear. The brick around the columns must be looked after at the fire side at least as often as shut-downs occur. The penalty for allowing boilers to sink is likely to be split pipe flanges if not worse. Fire-tube boilers lose their proper mud drainage by uneven settling and may develop leaky blowoff flanges as well.

Item (e) means the observation of pipe line behavior at all times, with, preferably, the trying out of pipe drains, automatic valves, etc., regularly. The penalty for leaving these things unobserved and unserviceable may be serious engine trouble of the most violent kind.

In special instances fire-tube boilers develop excessive corrosion externally on the rear tube sheet or where the combustion gases enter the tubes. This is due to the formation of an active acid coming from the burning of the sulphur of the coal combining with excessive moisture. The moisture may come from ashpit exhaust steam, or wasteful use of steam in jets, or leaky flues right at the place of corrosion. Soot depositing on the sheet is baked to the metal and absorbs the acid where it is permitted free action.

This trouble is minimized by soot prevention and removal, the avoidance of excessive moisture and improvement of the combustion process so that higher temperatures are the usual condition. External inspection at shut-down periods rarely reveals the degree of corrosion unless progressed too far for safety; gaging the sheet thickness carefully when renewing tubes affords the best chance, for then the difference between the sheet thickness as corroded and at the edges of tube holes (protected against corrosion by the tube bead) can readily be observed. A thick layer of baked scale lying tight against metal in gas of moderate temperature is one of the most active corrosive agents encountered in boiler practice.

Economy and Capacity

Main causes for lowered economy are not many in number, but the number of ways in which these causes distribute themselves are numerous and are considered in detail.

Incomplete Combustion by Losses to the Ashpit

Such losses by the rejection of partially burned fuel are in some cases very large. The best method to pursue to determine them is to require the measuring of the refuse resulting from all the fuel burned, locating on the record the amount against each boiler unit or furnace; placing the quantity alongside of the weight which should result if only the ash, as given by a fuel analysis, were present, a comparison of these items will give fairly accurate information as to the amount of burnable fuel going to the ash heap. If it is necessary to wet down the ashes and depend upon measuring, the weight of a dry barrow of ashes will serve as a unit weight for all but the most accurate cases. If the weight so determined runs steadily more than 25 per cent of the true ash content, then a search should be made for the causes. These are (1) faulty grates, as with broken down plain grates, or burned stoker bars or chain grate links; (2) insufficient draft in the furnace, compelling "puddling" of the fire to get air through thus making masses of clinker, the removal of which entails an excessive removal of good fuel; (3) coal of a character unsuited to the grate or stoker as, for instance, a coking coal on a chain grate stoker.

This inspection service becomes onerous where many boilers are operated because each unit ought to be separately handled so as to show how each grate behaves; also, different firemen will cause different losses and there ought to be some way to check one crew against another. Necessarily, the weight of fuel fired must be known to the same degree of accuracy as the ash weight. He who is earnestly striving to accomplish economy in fuel burning will devise ways to obtain continuous records; for the encouragement of the engineer who is forced to operate without modern weighing devices, it may be said that when once a weighing system is employed those who buy the coal will see the value of purchasing the equipment that calls for less effort on the part of the operatives.

Remedies for the first fault are obvious. When insufficient draft forces over-manipulation of the fuel bed, the remedies can only be found by exhaustive search for construction faults which are in the nature of wrong bridgewalls, arches, uptakes, damper areas, breechings, soot accumulations, baffles, air leaks and, in too many cases, too little chimney draft to begin with. To attempt to offer a single corrective would be foolish; until the trouble is located, the only salvation is the manipulation of the fire with minimum fuel bed thickness. A symptom of the trouble is exceedingly high furnace temperatures without corresponding high rate of combustion, a "lazy" gas movement from the furnace usually styled "bottled up heat."

The engineer should consult the stoker manufacturer concerning the third class of ashpit loss, the end in view being the selection of the best (not necessarily the most expensive) size and character of fuel the market affords adaptable to the grate equipment.

Incomplete Combustion and Gases Partially Burned

To determine whether or not gases are completely burned, the use of a gas analysis outfit is essential but this does not necessarily involve complicated or expensive equipment. A few words of caution are worth while, as many engineers deceive themselves by proceeding on the wrong basis and then are disappointed at the results. First: Take fair or average samples as close to the origin of the gas as possible so as not to be upset by the air which enters the setting after the combustion process has been completed as far as it can proceed. Be sure that the samples are taken in places where flame action has ceased.

Second: Do not strive to get more than 12 per cent CO_2 with hand-fired furnaces or 14 per cent CO_2 with stoker-fired furnaces. The penalty for too high CO_2 may be, and very probably will be, incomplete combustion.

Third: Endeavor to determine not only the CO_2, but

the percentages of oxygen and carbon-monoxide (CO) also. If CO occurs frequently even to a small part of one per cent the loss is large. Aim to burn the fuel with as small a quantity of air as possible without encountering incomplete combustion.

Incomplete Combustion and Smoke

A visible exhalation from the top of a stack is an indication that either ash or unburned carbon is passing into the atmosphere. The technique of smoke prevention, while simple enough in principle, is somewhat involved when all of the features are considered. Sufficient to say, if the plant runs so as to have complete combustion of invisible gases, the very means taken to bring about this result will be effective in the prevention of visible smoke.

Excess Air Introduced in Incorrect Quantities and Ways Through the Fuel Bed or Introduced in Incorrect Quantities and Ways Over the Fire

Too much air is just as uneconomical as too little, and it also happens that the former fault is the more prevalent. The remedies for both lie in the same place, namely, improved fuel bed construction combined with manipulation of draft in furnace.

For instance, should it be found that the resulting CO_2 is low, say 7 per cent, then there are either holes in the fire, the general fuel bed is too thin, or air admission over the fire is excessive. Uncovered grates are always productive of excessive air; chain grates burning "short" and the like are instances of special importance. Excess air over the fire, except during a very short interval immediately after firing, usually results in a like indication and fuel loss.

Even with a uniform fuel bed excess air may still result, in which case recourse to two temporary expedients may be taken; either the fuel bed may be increased in thickness or the draft may be reduced by dampering down. The first alternative should be tried first, but not to the extent of producing excessive clinker. It is better to operate with high furnace draft due to increased active fuel bed resistance than to reduce the furnace draft to overcome excess air passing through the furnace. The usual experience then is to produce more steam than needed; such a result simply confirms the inspection and the ultimate remedy is to reduce the amount of grate in service. Where several grates are employed at the same time, whole units may be dropped; where few or only one are used, a reduction in the size of each is feasible. The engineer who effects a grate reduction by either method is on the straight road to greater economy.

Excess Air Added Through Setting Cracks by Infiltration

When two gas analyses made upon the same boiler setting, one close to the fire and the other at the boiler setting outlet, firing conditions remaining the same, show an appreciable difference in CO_2 percentage or oxygen percentage the cause must be due to one of two things or a combination of both, namely, the taking of non-representative gas samples, or air infiltration. More obvious and simpler means of locating leakage cracks are by the use of candle flames or smoking waste, but such tests will give no idea of the amount of damage done to economy.

Losses entailed by air leaks are truly enormous, in many instances being great enough to lower uptake temperatures so as to deceive the engineer into believing that excellent heat absorption is taking place. Aside from this source of loss is that of reducing the available furnace draft, possibly to the extent where an additional grate must be fired in order to burn sufficient coal to carry the load. Places to look for such trouble are many and the best time to investigate is during operation, after all boiler and setting parts have reached their greatest expansions. The principal ones are:

(a) General brick walls. Cover these with any one of the many air-leak preventives after calking the largest cracks with asbestos rope or wicking soaked in the same preparation.

(b) Hollow staybolts of water-tube boilers. Plug them up.

(c) Where metal side cleaning door frames of water-tube boilers join with brickwork.

(d) Around and between water-tube boiler headers.

(e) Between back arches and rear tube sheets of fire-tube boilers.

(f) Along the sides of return tubular boilers where the brickwork meets the shell.

(g) Around drums projecting through brickwork as with water-tube boilers.

(h) Around cleanout doors, inspection doors, soot blower access pipe openings, blowoff piping.

The gains obtained by correcting setting air leaks are indicated by the changes shown in either better gas analyses at the boiler outlets, higher uptake temperatures or less draft losses within the settings.

Necessarily, where piping passes through masonry walls, the material used to stop up the cracks should be pliable; mineral wool or asbestos in its several forms offers a variety of choices; the main thing is to stop the leak without subjecting any pressure part to injurious stresses.

Dirty Heating Surface on the Interior or Water Side

Under this head some discussion has already been offered on the subject of safety.

Gradual building up of uptake temperatures without corresponding other causes is a quite reliable index of growing scale thickness. Losses due to scale are in exact proportion to such temperature increases, other things remaining the same. In the case of scale, economy and capacity travel in like directions, that which improves one having a good effect on the other.

Dirty Exterior Heating Surface From Soot, Etc.

Soot is a far better insulator against heat transmission than a corresponding thickness of scale, besides offering in many places an excellent medium for the absorption of corrosive acids. The only answer is that soot is needless and its removal is imperative if any degree of economy is to be gained.

Shut-down inspection ought to be religiously employed to discover soot deposits. It is particularly at

these times that the daily soot blowing effect can be studied as by finding large accumulations driven into, or at least not dislodged from boiler spaces by steam jet blowers.

Service inspections are really of three kinds directed to the same end. The first is to see to it that the blowing is done thoroughly by actual observation of the labor, not depending upon the report of the helpers to that effect. It should be required to be done not only at some time during low capacity operation, but most particularly when the general temperatures are highest as during high capacity. A good time is right after the noon shut-down whistle blows, thus giving at least one useful outlet for excess steam pressure when the power load goes off.

Occasional extra soot blowing not at predetermined times, noting the fall of uptake temperature, constitutes a second check. If the temperature reduction is appreciable, it is ample evidence that either the previous blowing was not well done or the blowings are too far apart.

A third method is to watch the record of a recording uptake thermometer; if such is not available, the same result to a less extent can be gained by means of an indicating instrument frequently read.

It is not extraordinary to encounter a reduction of 75 deg. F. due to blowing soot from tubes. An appreciable increase in economy will result where the removal of soot effects such a reduction. By any and every means give the laborers decent soot-blowing equipment to work with, as no good man will stay with an inefficient blowing outfit any longer than it takes to get employment elsewhere. Soot accumulation is another one of the causes for a gradual increase of setting-draft loss.

Steam Leaks

All leaks betray their presence by the noise they make.

Blowoff cocks and valves are a prolific source of steam leaks. Feel them an hour after blowing down and if they are hot, a leak exists. Also watch the drip and the rising vapor coming from the end of the blowoff line. Steam traps that fail to "trap" are worse than cracked valves; it is not difficult to find out whether they work. Traps are among the neglected equipment around a steam plant that is usually hidden away where it is not nursed as it should be; inspect them every watch, and fix them if they go wrong.

Where leaks occur at faulty calked joints or seams of boilers and at poorly rolled tube ends, they are revealed by the noise of escaping steam and water. A noise of this kind justifies an immediate shut-down, as costly damage may result to the boiler sheets from the cutting action of the steam jet. Similar faults occurring under thick insulation show up by producing a mushy patch near the leaky place.

Excessive Radiation

Where heat is to be kept either in or out, insulation in some form pays. It is not necessary to insulate steam lines used only during short periods at intervals several hours apart, provided the control valve is close to the steam supply; branch lines serving soot-blower elements are cases in point.

Boiler heads ought to be well covered and later looked after to see that the application is not loosened; a loose insulation allowing vertical travel of heated air (a chimney so to speak) is worse than no insulation at all.

Coverings for brick setting pay, more because they form an air-infiltration preventive than because of the prevention of heat radiation, but the combined advantages, if the work is well done, usually result in good interest on the investment. It doesn't follow, however, that expensively lined sheet steel coverings are the best way to overcome heat radiation; they sometimes cover up or hide from view serious air leakage cracks. It is the inside wall that performs the biggest part of the business of holding in the heat and preventing the air from seeping into the setting. Inspection simply means that these things must be looked after whenever faults occur.

Gradual Deterioration

Temperature readings at boiler uptakes showing unaccountable increases may be understood when a shut-down inspection reveals the displacement of baffles, thus allowing a short-circuit for the gases on their way to the breeching. Such cases occur frequently especially where tube baffles are supposed to meet brick wall constructions. Draft readings indicate the same faults, and often it is possible to see the gases carrying sparks going in directions not intended. Baffles are often displaced by the spring of sootblower elements rearing back on account of the reaction set up at the blower nozzles.

Where brickwork in furnaces fails after giving a good term of service there is nothing to do but replace it. There are, however, cases where arches come down without any particular reason and when replaced repeat the performance. In such cases an inspection of the brickwork holding the skewbacks in place had better be made; in addition, rigid supporting irons in line with the skewback thrust should be installed. The failure is really due to the insecure masonry holding the skew in place rather than to poor workmanship and material in the arch itself.

Design Improvement

Remarks that go before are tantamount to saying, learn by constant investigation the possibilities of your furnace-boilers as units, noting the effects of this or that operating method, not by the insecure evidence of short boiler trials, but by the solid unmistakable evidence of continuous records of coal consumption supported by routine and special gas, draft and temperature records. It is presumed that the enthusiasm awakened by the good effects of routine changes such as effective soot blowing and stopping of leaks of all kinds will urge more ambitious reformations.

Common experience of extended study in boiler rooms is to find that draft is inadequate in the furnaces, grate areas are out of proportion to steam demands, labor for handling coal and ashes is too costly, the kind of stoking equipment is unadapted to the most available coal size or quality, or a combination of some

of these puts the plant in an unresponsive condition respecting changes of load. Now comes the time when draft analyses include a study of all items entering into its production in the furnaces.

Absolutely no rule is possible to state except that the plotting of draft data, thus giving a gradient of the entire system starting at the chimney extending to the furnaces, is essential. From such a plotting, the relative effects of this or that change of shape or size of opening, the effect of breeching insulation, the enlargement of damper openings, the easing of breeching turns and similar things can be considered. Places to look for the most serious draft deficiencies are:

1. Draft creator or chimney not high enough.
2. Too sharp a turn of breeching into the chimney.
3. A bad change of shape where breeching and chimney meet.
4. Breeching admits air or radiates too much heat.
5. Breeching, if underground, evaporates water seepage or is continually wet.
6. Dampers of boiler outlets stand in the way of oncoming gases.
7. Damper openings are too small or badly shaped.
8. Gases from different boilers meet head on.
9. Breeching damper, without any function, stands in the way.
10. Breeching has sharp turns or is made with one dimension in section much larger than the other.
11. Damper openings not as wide as the settings.
12. Circulating tubes of boilers stand in front of damper openings.
13. Damper plates placed so that no position permits full opening.
14. Soot and air seepage in settings.
15. Superheater elements fill up passes too much.
16. Boiler baffles incorrectly located.
17. Furnace brickwork too restricted.
18. Bridgewalls too high.
19. Back arches of fire tube boiler settings too near tube sheets.
20. Air inlets to ashpits too small.
21. Air cannot enter boiler room. (Frequent in winter.)

The question of grate area is so involved with available draft, the character of grate equipment, furnace construction and the permissible rates at which units may or ought to be operated that it is futile to try to lay down universal rules. Before coming to conclusions it is, however, necessary to be sure that the air spaces are free and that the parts are intact in whatever grate is considered. Of course, when sizing up draft matters the grates ought to be in normal condition or the draft readings will be distorted. Whether any extensive investigations are made or not, there is no excuse for neglecting the grate equipment.

Finally, the engineer who confines his inspection work to his own plant is denying himself some of the best information available. He should make a practice of calling upon his brothers in the same line of endeavor and profit by their experiences.

Keeping Track of Boiler Performance

WEIGHING COAL, METERING FEED WATER, MEASURING DRAFT, ANALYZING FLUE GAS, WORKING DATA INTO USABLE FORM. BY H. A. WILCOX

THE VITAL problem confronting the operating engineer of today is how to keep track of what his boilers are doing and to maintain them at all times at their best efficiency.

In order to know what his boilers are doing, the engineer must know, first, how much coal they are consuming in any given period of time and, second, how much water is evaporated into steam at boiler pressure by the heat produced by burning this given amount of coal.

There are numerous ways of ascertaining the amount of coal used and I will simply suggest a few from which each reader may choose the method best suited to his individual conditions.

1. For large, well equipped plants, automatic weighing chutes or hoppers are generally used. These vary in type and style and are quite accurate. There are a number of good makes on the market today to select from.

2. Some plants use charging cars. These cars are loaded outside the boiler room and are pushed onto large platform scales. The net weight of coal in the car is recorded and the coal is then left in the car; the car pushed into the boiler room, and the fireman shovels directly from the car to the boiler until the car is empty, when it is switched out for reloading. Sometimes a type-registering beam is used on the scale. This obviates the errors and difficulties with regard to the inability of the coal handlers to read and write, as the weights are printed upon tickets and the tickets dropped into a box at each weighing. Different colored tickets may be used to differentiate between hard and soft coal in plants where a mixture is used.

3. This method is exactly like the second except that wheelbarrows are used in place of cars and the coal is dumped on the boiler room floor.

4. This method or a modification of it is the poorest of all and is in use in most all plants today. Coal is brought into the boiler room in wheelbarrow or tip cart loads and dumped on the floor. Once a day, week or month one or more loads are weighed. Then the number of loads per shift multiplied by this standard weight is supposed to be the coal used on that shift. Anyone of ordinary intelligence may readily see where large errors are probable, so I will not dwell upon this item.

5. The inventory method. This method may be used in combination with any of the others for obtaining a check upon them and may be used to obtain the weekly, monthly or yearly coal consumption. The entire quantity of coal on hand at the plant is measured on the first day of any month. This measurement should be made

in cubic feet. Reduce this measurement to tons. As the volume and weight of coal varies for different localities, I will simply state that 43 cu. ft. of bituminous coal per long ton is an average figure. Experimental weighing of a box of coal holding just one cubic foot will determine this figure for any coal and it should be checked up periodically. On the first day of the next month a similar estimate of coal on hand should be made and be reduced to tons. Then the amount on hand at the first measurement plus the amount received between the two measurements (this will be accurate, as it is the figure upon which the coal is paid for) and less the amount on hand at the second measurement will equal the amount consumed between the two measurements. An error in measurement one way or the other will be offset by errors in subsequent measurements and thus large cumulative errors cannot occur as in the case of the wheelbarrow method.

Very little choice can be given for measuring the water fed to the boilers over any considerable period of time, as it is practically up to the reader to choose for himself between the merits of the numerous hot water meters now upon the market, measuring by meter being the only practical way to do this.

Rate of combustion, or pounds of coal burned per square foot of grate per hour may easily be calculated at any time by dividing the pounds of coal burned during any period of time by the entire grate surface in use multiplied by the period of time in hours. In general, it may be stated that a small grate surface and a high rate of combustion are desirable. However, variable conditions in regard to coal, draft, etc., influence these items so much that it is impossible to make any set rule for all plants. Good practice, with natural draft, for obtaining the best efficiency, is about 13 lb. of anthracite coal burned per square foot of grate surface per hour and 24 lb. of bituminous. These rates may be very greatly increased by the application of forced draft if necessary.

Evaporation per pound of coal is the true measure of a boiler's performance and will indicate whether a boiler is doing good work or not. This figure is obtained by dividing the pounds of water fed to the boiler by the pounds of coal burned under the boiler during the same period of time. As feed water temperatures, etc., vary in different plants, it is customary for purposes of comparison to refer all measures of evaporation to a standard called "Equivalent Evaporation." This means the evaporation of a certain quantity of water from and at 212 deg. F. into steam at atmospheric pressure. Given the actual pounds of water fed to a boiler, the temperature of the feed water, and the steam pressure, the equivalent vaporation may be found by multiplying the pounds of feed water by a factor called the factor of evaporation. This factor may be found in tables in a number of standard handbooks, under the corresponding feed water temperature and steam pressure. There is an exact method for calculating the factor of evaporation, but the foregoing method is good enough for all practical purposes. The equivalent evaporation, divided by the pounds of coal burned, may then be compared with the results of standard practice. It is generally conceded that an equivalent evaporation per pound of coal fired, of from 8 to 12 lb. constitutes average continuous practice in ordinary plants. With coal averaging 14,500 to 15,000 B.t.u. as received and a combined boiler and furnace efficiency of 75 per cent, an evaporation of between 11 and 12 lb. would be expected, while coal having only 12,000 B.t.u. and a boiler and furnace efficiency of 65 per cent would produce an evaporation of about 8 lb.. Much depends upon the efficiency of the operating force, the equipment, draft, type of boiler, load factor, etc.

The importance of draft gage readings does not seem to be fully appreciated at the present time in most plants. This is probably due to the fact that there are a great many variables which govern the draft; such as thickness of fuel bed, height of chimney, grade of coal, climatic conditions, etc. It is assumed that my readers are already familiar with the ordinary methods of measuring draft by means of the U tube. Ordinarily draft gage readings are taken in the ash-pit, over the fire, in the breeching, and perhaps at the base of the chimney. Of all these points, the draft over the fire is the most important as it is generally true, that with a good grade of bituminous coal a draft of at least 0.25 in. (inches of water) is necessary if 100 per cent rating is to be obtained from the boiler with natural draft. As stated previously, draft is dependent among other things upon the rate of combustion, and experiments have been made which indicate that the draft at the fuel bed varies from 0.07 in. for a combustion rate of 10 lb. of coal per square foot of grate surface per hour with run of mine of coal per square foot of grate surface per hour with run of mine bituminous coal to 1.37 in. for a combustion rate of 50 lb. with semi-bituminous slack. The full results of the experiments referred to have been printed in tabular form and are available in the handbooks. Draft readings in the breeching and at the stack are valuable to indicate the draft losses in the boiler and flue. The United States government, in a series of tests, has demonstrated that if the fuel bed is doubled in thickness, in order to force the same weight of gas through it, the pressure drop through the fuel bed must be doubled. Similarly, to quadruple the thickness of the fuel bed, makes it necessary to quadruple the pressure drop. This same law also applies to either increasing or decreasing the cross sectional area of the gas passages throughout the boiler. This law is important on account of the fact that it governs combustion rates, changes in the baffling and other interior changes in the boiler.

Differential draft gages are instruments so constructed that the pressure drop between any two points in the boiler may be readily measured. For instance, one side of the instrument is connected to the ash-pit and the other side to the space over the fire. The reading of the instrument gives the total pressure drop through the fuel bed. To illustrate: Suppose a boiler steaming at 100 per cent rating has a total pressure drop of 2 in. and requires a fuel bed thickness of 10 in. for its operation. Suppose the differential draft gage readings show a pressure drop of 1.2 in. through the fuel bed and 0.8 in. pressure drop through the remainder of the boiler.

For some reason, it becomes necessary to force this boiler to 150 per cent rating. This will mean roughly to increase the rate of combustion and consequently the weight of gases passing through the boiler by 50 per cent. To accomplish this result, the resistance to the flow of the gases through the boiler must be lessened somewhere, or the total pressure drop must be increased. Changes in the baffling or setting would be inadvisable. Therefore the only way is to reduce the thickness of the fuel bed. A pressure drop, above the fuel bed, 50 per cent greater will be required. Therefore this must be made $(1.50\times.8)=1.2$ in. If the total available pressure drop is only 2 in., this will leave only 0.8 in. for the pressure drop through the fuel bed. Therefore the fuel bed thickness must be reduced to $\frac{10\times.8}{1.2\times1.50}=24.44$ in.

This of course would mean very frequent firing and holes in the fire would quickly develop. Therefore in actual practice the pressure drop would probably be increased through the application of some form of forced or induced draft. The foregoing rule holds true at constant temperatures and a use of this rule will indicate the approximate result to be expected from any change in fuel bed thickness, baffling, etc. In recent years balanced draft systems have become quite popular. The general equipment of such systems consists of a blower, the speed of which is automatically regulated so that the draft over the fire is maintained at about atmospheric pressure at all times. The following advantages are derived: (1) The velocity of the gases through the boiler is reduced thereby allowing time for more complete combustion and for better absorption of the heat of the gases by the heating surfaces of the boiler. (2) Infiltration of air through the setting and short circuits in the baffling are eliminated due to the balanced pressures. (3) Load fluctuations are readily taken care of. (4) The correct proportion of air to fuel is more easily maintained. This element is of considerable importance as will be shown in the succeeding paragraphs which take up the question of flue gas analysis.

Hand in hand with draft gages come CO_2 (carbon dioxide) indicators. There are three distinct types of these machines on the market. The continuous CO_2 machine which automatically and continuously analyzes and records the per cent of CO_2, only, in the flue gases. The hand apparatus or Orsat with which it is possible to obtain the percent of CO_2, O (oxygen) and CO (carbon monoxide) in the flue gases at any time and point. The small hand machine with which it is possible to obtain the per cent of CO_2, only, in the flue gases at any time and point. There are numerous treatises on the care and operation of these machines and the methods of obtaining samples of gas for analysis and the reader is respectfully referred to them as lack of space prevents me from going into these details. My personal preference is for the Orsat as it gives a more complete analysis of the gases; can draw gas samples from different boilers and from different points in the same boiler; the chances for air leaks in the apparatus are less, and they are less complicated. A CO_2 recorder kept in continuous operation is of great value for comparing the work of different firemen and detecting poor firing methods.

The next question is, "What do the results of flue gas analyses indicate?" For practical purposes, the combustible of coal is chiefly carbon and hydrogen. When these two components burn, they unite with the oxygen of the air to produce CO_2 (carbon dioxide) and water vapor. It has been estimated that to obtain complete combustion of the carbon in coal about 40 per cent of excess air must be supplied over the theoretical amount needed. Incomplete combustion of the carbon in the coal produces carbon monoxide (CO). We therefore analyze for CO_2, O and CO as these three gases indicate the true process of combustion of coal. Ordinarily the percentage of CO in a sample of flue gas is very small and the higher the CO_2 the lower the O and vice versa.

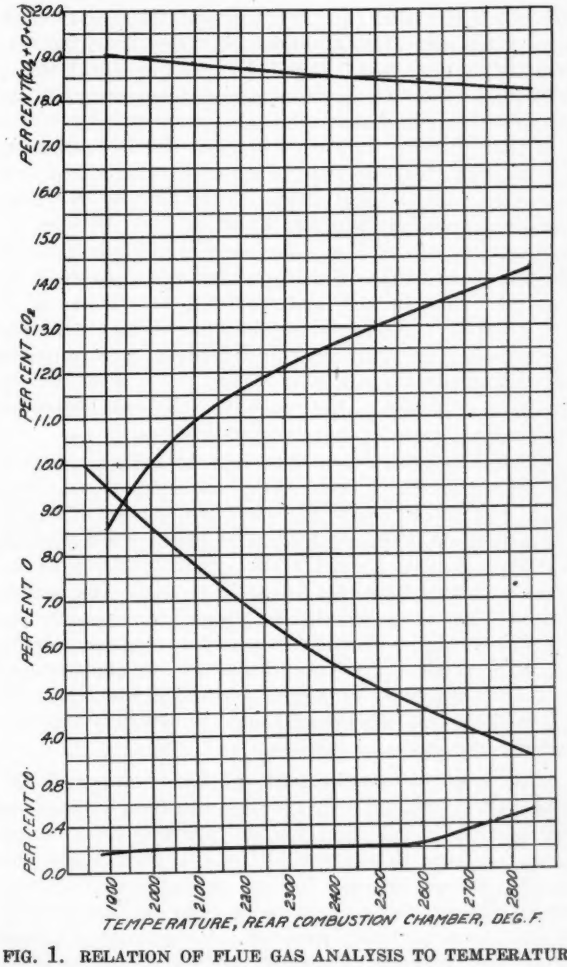


FIG. 1. RELATION OF FLUE GAS ANALYSIS TO TEMPERATURE IN REAR COMBUSTION CHAMBER

Good results for CO_2 range from 10 to 15 per cent in a given sample of flue gas and results above or below these figures indicate trouble somewhere as a general thing. Below 10 per cent usually indicates too great a supply of excess air, holes in the fire, leaky settings, or trouble in the Orsat itself. Above 15 per cent usually indicates a deficiency of air, and is generally accompanied

by a rise in the per cent of CO which means incomplete combustion.

By analyzing samples of gas taken from various points in the gas passages, we may keep tab on the progress of combustion throughout the boiler. As complete combustion produces the highest temperature, most of our attention should be concentrated on the rear combustion chamber, as it is here that we want the highest temperature or just before the gases come in contact with the heating surfaces of the boiler. The two curves shown herewith are self-explanatory. They were obtained from a report of the United States Bureau of Mines which covers the results of about 500 tests made some years ago. Gas samples were taken from the rear combustion chambers. It will be noted in Fig. 1 that a high percent of CO_2 produces a high temperature in the rear combustion chamber but that the per cent of CO also rises sharply, indicating a sharp rise in incomplete combustion. Therefore it is apparent that there is a minimum value beyond which it does not pay to reduce the air supply. It will also be noted that the curve of totals at the top of Fig. 1 decreases with an increase in the per cent of CO_2. This drop may be due to the presence of hydrogen and hydrocarbons, but its reason is not generally understood at present. I may say that the results of my own work in different plants conforms very closely to the curves in these figures, although my totals of CO_2, CO and O have generally run somewhat higher than the total curve in Fig. 1.

There is another source of loss in boiler operation which can be easily detected by the engineer. I refer to the per cent of combustible left in the ashes. A careful inspection of the ash pile, after the fires are cleaned, will often disclose a considerable quantity of coke in the ashes. This is the result of careless cleaning or broken grates. The ashes should be sampled occasionally by the quartering method (as applied to coal sampling) and an analysis of these samples by a competent chemist will disclose the percentage of loss through combustible left in the ashes. This point should be carefully watched and the loss kept at a minimum at all times.

The engineer should take the temperature of the gases, leaving the boiler frequently. In fact, a recording thermometer in the main flue will prove to be a paying investment for this purpose. In a perfect boiler the temperature of the gases leaving the boiler would never be more than a few degrees higher than the temperature of the steam at boiler pressure. Unfortunately, there is no such thing as a perfect boiler and we therefore estimate that a flue gas temperature about 100 deg. higher than that of the steam at boiler pressure is unavoidable. If the engineer finds that this temperature is 200 to 300 deg. higher than the steam temperature, he may be fairly certain that it is caused by one of three things: 1. Forcing the boiler very hard and consequently raising the velocity of the gases through the boiler to a high rate. 2. Slow or incomplete combustion in the furnace. That is, combustion is still going on after the gases leave the boiler. 3. There are short circuits in the gas passages or baffling in the boiler.

In any case, excessive flue gas temperatures indicate a decided loss and the cause should be ascertained and remedied at once if possible. Where limited capacity makes forcing the boiler necessary, a part of this loss can be prevented by the installation of a suitable feed water economizer, so that the heat in the flue gases will be imparted to the feed water instead of being lost up the stack.

In the course of this article I have several times referred to "Boiler Efficiency" and I will now try to define it briefly:

Efficiency of boiler and grate equals heat absorbed by boiler per pound coal as fire divided by calorific value of one pound of coal as fire.

I have already shown, earlier in this article, how to obtain the equivalent evaporation per pound of coal as fired. Then the equivalent evaporation, multiplied by the familiar factor 970.4, represents the heat absorbed by the boiler per pound of coal as fired. An analysis of the coal will give the B.t.u. per pound of coal as received and this represents the calorific value of one pound of coal as fired. Substitute in the formula above and the result is the boiler and grate efficiency. Example:

Water fed to boiler 86,000 lb.
Coal actually fired 10,000 lb.
Feed water temperature 161.8 deg. F.
Steam pressure (gage) 151.0 lb.
Barometer 28.5 in. mercury.
Quality of steam (assumed) 98 per cent.
B.t.u. per pound of coal as received 11,250.

$$\text{Actual evaporation} = \frac{86{,}000}{10{,}000} = 8.60.$$

Factor of evaporation (from tables) = 1.08.

Equivalent evaporation per pound of coal as fired $8.60 \times 1.08 = 9.288$.

Heat absorbed by the boiler per pound of coal as fired $(9.288 \times 970.4) = 9.013$ B.t.u.

$$\text{Efficiency of boiler and grate} \frac{9.013}{11{,}250} \times 100 = 80.11$$

per cent.

The builder of a boiler usually furnishes an efficiency curve showing the efficiencies to be expected from the boiler under various conditions. Let the engineer figure out the efficiency of the boiler and grate as often as possible and plot a curve under similar conditions to that of the curve supplied by the boiler builder. If the builder's curve is better than that obtained by the engineer, something is generally wrong somewhere. It may be that the builder's curve applies to the boiler alone. In this case the engineer can get a similar figure for the boiler by using the formula:

Efficiency of boiler alone equals heat absorbed by the boiler per pound of combustible burned on the grate divided by calorific value of one pound of combustible as fired.

This formula is worked exactly like the preceding one except that the per cent of combustible in the coal must be obtained from the analysis and the value "pounds of coal as fired" modified by this percentage.

Efficiency of grate alone equals efficiency of boiler and grade divided by efficiency of boiler alone.

It would be well to figure the efficiency of the grate alone and see that this is up to the proper standard.

If the engineer is still unable to locate a reason for low efficiency, he can then figure out a complete heat balance for the boiler, an intelligent study of which is almost certain to locate the trouble, and he can then "go to it."

Having obtained a complete heat balance, all the conditions of the operation of the boiler are thereby brought out and its case may be studied as carefully as a doctor studies the symptoms of a sick person. After all, when you stop to think of it, the operation of a boiler is very much like the ordinary working of human life, and a good doctor does not give medicine until he knows where the trouble is and has some idea at least of its cause. Then he combats the disease with every known aid at his command. In case of doubt, he calls in the expert. A boiler eats and drinks just the same as a person does and it has to have these things administered to it in the proper way if it is to do good work.

I consider the heat balance of so much importance

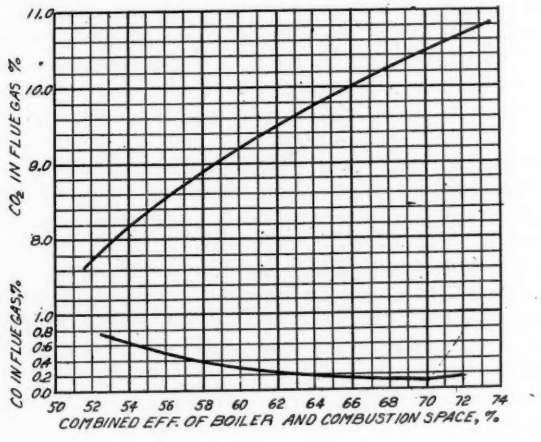


FIG. 2. RELATION BETWEEN COMBINED BOILER AND FURNACE EFFICIENCY AND FLUE GAS ANALYSES

in keeping track of boiler performance that I have deemed it fitting to conclude this article with a simple method for working out a heat balance for a boiler, together with a practical example of its use. Of course all plants have not the facilities for obtaining all the information necessary for working out a complete heat balance, but by assuming a few quantities such as barometer, quality and, if necessary, getting the ultimate coal analysis by averaging some of the government analyses of coal similar to that in use at the plant, a rough idea of the efficiency of the boiler and grate may be obtained, and this will be worth while.

Heat Balance for a Boiler

Item and symbol—	B.t.u.	%	$
Heat absorbed by the boiler.....E			
Dry flue gas loss.....L_1			
Unburned carbon loss.....L_2			
Loss from moisture in coal.....L_3			
Loss from burning hydrogen.....L_4			
Radiation and unaccounted for.....L_5			
B.t.u. com. and coal bill.....		100	

Heat Balance Data

Total coal pounds, B.
Total water pounds, J.
Average CO_2, G.
Average stack temperature, T.
Actual evaporation, J/B.
Average steam pressure gage, P.
Average feed water temperature, F.
B.t.u. in one pound of steam at P + barometer and at F, D.
Ash commercial, A.
Unburned carbon, Y.
Moisture, M (in coal).
Total carbon in the coal, C.
Hydrogen in the coal, H.
Nitrogen in the coal, N.
Total dry products of combustion per pound of coal, W.
Temperature of the air entering the furnace, t.

Instructions for Figuring the Heat Balance

$$E = J/B \times D$$

$$L_1 = W \times 0.237\ (T - 70)$$

$$L_2 = \frac{145 \times A \times Y}{100 - Y}$$

$$L_3 = \frac{M}{100}\ (1182.4 - t) + 0.46\ (T - 212)$$

$$L_4 = 9 \times \frac{H}{100}\ (1182.4 - t) + 0.46\ (T - 212)$$

L_5 = B.t.u. commercial $(E + L_1 + L_2 + L_3 + L_4)$.

Practical example worked out.

B—480,000 lb.	Y—18.67 per cent.
J—4,290,000 lb.	M—3.60 per cent.
G—10 per cent.	C—78.76 per cent.
T—570 deg. F.	H—4.47 per cent.
P—125.0 lb. gage.	N—1.35 per cent.
F—192.8 deg. F.	t—70 deg. F.
A—7.21 per cent.	B.t.u. commercial—14,148.

Barometer—14.7 lb. per sq. in.

Three things must be computed—J/B, D, and W.

$$J/B = \frac{4{,}290{,}000}{480{,}000} = 8.94 \text{ lb. of water per lb. of coal.}$$

To compute D, it is necessary to express P in terms of absolute pressure: $P + 14.7 = 125.0 + 14.7 = 139.7$ lb. per sq. in. absolute. $F = 192.8$ deg. F.

Turn to steam tables and find "Total Heat of Steam" corresponding to a pressure of 139.7 lb. Answer—1192.2 B.t.u.

Turn to steam tables and find "Heat of the Liquid" corresponding to a temperature of 192.8 deg. F. Answer—160.7 B.t.u.

Then $D = 1192.2 - 160.7 = 1031.5$ B.t.u. In this example we have considered the quality of the steam to be 100 deg.

If it had been 98 per cent we would have multiplied the "Total Heat of Steam" by 98 per cent before subtracting the "Heat of Liquid."

$$W = \frac{C}{100} + \frac{243.6 \times C}{G \times 100} = \frac{78.76}{100} + \frac{243.6 \times 78.76}{10 \times 100} =$$

$.7876 + 19.18 = 19.97.$

$E = 8.94 \times 1031.5 =$ ..... 9224 B.t.u.

$L_1 = 19.97 \times 0.237\ (570 - 70) =$ ..... 2367 "

$$L_2 = \frac{145 \times 7.21 \times 18.67}{100 - 18.67} = \text{.....} \quad 240 \text{ "}$$

$L_3 = 0.360\ (1182.4 - 70) + 0.46\ (570 - 212)$ ..... 46 B.t.u.

$L_4 = 9 \times 0.447\ (1182.4 - 70) + 0.46\ (570 - 212)$ ..... 514 "

$L_5 = 14{,}148 - (9224 + 2367 + 240 + 46 + 514)$ ..... 1757 "

14,148 B.t.u.

If the reader wishes to see what these figures are in per cent, divide each one of the B.t.u.'s by 14,148. The total of the per cents obtained this way should equal 100.

Boiler Equipment

RIGHT PROVISION FOR BOILER FEEDING, BLOWING DOWN, AND ENSURING SAFE OPERATION. BY MARTIN O'CONNOR

SAFE, CONTINUOUS and efficient operation of the boiler plant depends to a considerable extent upon the selection, arrangement and care of the numerous fittings and auxiliaries connected to the boiler. The object of this article will be to give the engineer a few suggestions along the above lines.

FEED WATER SUPPLY

FEED water pipe to the boiler should be fitted with a valve as close to the boiler as possible and between this valve and the pump a check valve. Where several boilers are supplied from the same feed line, a valve should be provided on each side of the check.

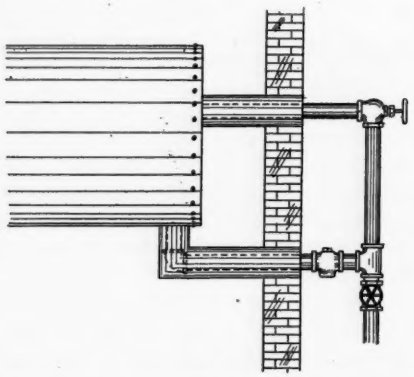


FIG. 1. COMBINATION SURFACE AND BOTTOM BLOWOFF WHICH MAY BE USED AS A CIRCULATING PIPE

The pipe should enter the boiler through a substantial flange in order to allow the proper number of threads, at least 4 for 1 and 1¼-in. piping, 5 for 1½ and 2-in. and 7 for 2½ to 4-in.

As a general rule, feed water should be discharged into a boiler where it is least liable to come in contact with surfaces exposed to the direct heat of the fire or deposit scale where it may be difficult to remove. The pipe should if possible extend for some distance after entering the boiler to allow the water to come nearly to the same temperature as the water in the boiler. A good arrangement for a tubular boiler is for the feed pipe to enter through the front head above the tubes and extend to within 3 ft. of the rear head and discharge below the water line.

Feed should be as continuous as possible while the boiler is in operation and proportional to the load so as to maintain a steady water line. Several good feed water regulators are on the market which will maintain an almost uniform water line under all variations of load. Regulators are divided into two general types: the float type in which the float, either operates a small pilot valve which in turn operates a diaphragm feed control valve, or the float may operate the control valve directly by a system of levers. The other, known as the expansion type, has an expansion device arranged to be affected by the height of the water and operate either a pilot valve and control valve as in the float type or a control valve direct by means of levers. Some regulators are combined with a high and low water alarm column, which makes a neat and compact arrangement but has the objection that any disarrangement of the float may render inoperative the low water alarm and also the regulator.

Where several boilers are fed by the same pump, a pump governor is essential to maintain the proper pressure in the feed line. Governors are of two types, one which maintains a constant pressure in the feed line regardless of the boiler pressure, and the excess pressure governor which, by means of differential diaphragms, carries the pressure in the feed line a certain predetermined amount above the boiler pressure.

All valves and fittings used on boilers, whether for steam or water, should be extra heavy if boiler pressure is to exceed 125 lb.

BLOWOFF

BLOWOFF pipe should be located at the lowest possible point of the water space in the boiler so as to remove all sediment possible by blowing off daily, or oftener if bad water is used. In large horizontal water-tube boilers two or more blowoff pipes are provided in the rear water leg or drum, as it is found that the sediment will flow but a short distance to the outlet when blowing off.

Where pipe is exposed to the heat of furnace gases, as in return tubular boiler settings, it should be protected by a fire brick pier built in front of the pipe or a removable sectional covering should be used. Where pipe is carried through setting wall, it should be surrounded by a section of larger pipe arranged to prevent any

strain on the blowoff by settling of the wall. The annular space between the pipes can be packed with asbestos rope to prevent the entrance of air.

Size of blowoff is limited by boiler rules in most states to 2½ in. All high-pressure boilers should have two valves in each blowoff leading from the boiler or a cock and valve, the cock to be placed next to the boiler. When blowing down, the cock or valve next the boiler is first opened wide, and closed last when through blowing off. This method brings all the wear on the outside valve which can be repaired while the boiler is in service. When the boiler is cleaned the cock should be taken apart and cleaned and a light application of oil and graphite will allow it to be turned easily.

Where more than one boiler is connected to the same blowoff line, many large boiler users provide a check valve between the valves and the line to prevent hot water backing up into an idle boiler and possibly causing injury to workmen. Surface blowoffs can be used to advantage where the water is muddy, and to be effective should be located as near the upper water line as possible and used frequently. A combination surface blowoff circulating pipe and bottom blowoff is shown in Fig. 1, which has been used to advantage in cases of very muddy water.

Water Columns

Water columns should be connected to boilers using as short piping as possible to bring the water gage in proper location where it can easily be seen by the attendant. Size of pipe should preferably be 1½-in., the connections from boiler to column to be fitted with crosses having brass clean-out plugs. Valves between boiler and column should be of the straight-way type and preferably have rising stems. These valves should be opened and closed once a week to keep the discs and seats free from scale. In some plants it may be advisable and in some states it is required to lock these valves open, using a small chain and padlock for the purpose.

Water columns should be blown down daily and occasionally the valve in steam connection should be closed and the column allowed to blow freely through the water connection. On tubular boilers, the column is set to bring the lower gage about 2½ in. above the top of the upper row of tubes. On water-tube boilers of the horizontal type the middle gage cock is usually set at the center line of the drum. Figure 2 shows arrangement of connections and blowoffs.

Steam Gage

Each boiler should have an accurate and reliable steam gage to be graduated to 1½ times the working pressure. It should be connected with the steam space of the boiler through a syphon or drop leg so as to keep the connection always filled with water. It should be substantially mounted where it will be free from vibration, which soon destroys a gage. If located any considerable distance below the connection to the boiler, allowance should be made for the increased pressure due to the water column at the rate of one pound for each 2.3 ft. height of water column.

Safety Valves

The A. S. M. E. Code requires two or more safety valves on each boiler except when the boiler is of such size that a 3-in. valve is ample. Where possible, the valves should be bolted directed to a separate flanged outlet on the boiler shell and in no case should any valve be placed between the safety valve and boiler or between the safety valve and the atmosphere. Where a discharge pipe leading to atmosphere is necessary, it should be full size of outlet or larger and should be fitted with an open drain arranged to prevent any water lodging in pipe or valve. The discharge pipe should be securely supported at the fitting next the outlet to take care of the heavy strain at this point when the valve is blowing.

The safety valve capacity for any boiler should be determined by assuming 6 lb. of water evaporated per square foot of boiler heating surface per hour. The discharge capacity can be obtained from the manufacturers and varies with different makes. Safety valves having outside springs are used with superheated steam

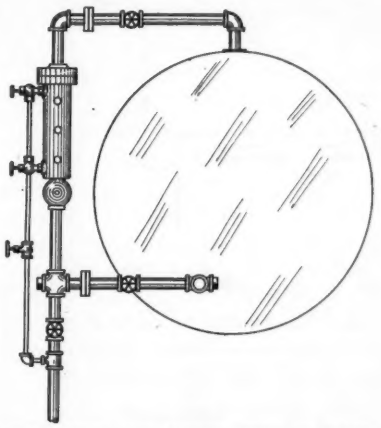


FIG. 2. ARRANGEMENT AND CONNECTION OF WATER COLUMN ON HORIZONTAL RETURN TUBULAR BOILER

to protect the spring from the high temperature. In operation all safety valves should be made to blow regularly, and any signs of leaking taken care of promptly by regrinding. Also the pressure at which the valve flows should be noted, and if not correct the valve should be adjusted. Lever safety valves are no longer allowed on new work.

Steam Outlet

Steam outlets on the tubular boilers were formerly taken from drums or domes under the impression that this was the proper way to obtain dry steam. This arrangement was later found to be a source of weakness to the boiler shell as well as furnishing additional surface for condensation. The internal dry pipe was found to be a better arrangement. This consists of an internal pipe located in the upper part of the steam space as shown in Fig. 3 and perforated on its upper half with ½-in. holes of a sufficient number so that their combined area shall be twice that of the steam outlet to which the dry pipe is connected. Several holes should be drilled in the bottom of the pipe to insure drainage.

This pipe should be examined when the boiler is cleaned and holes not allowed to fill with scale.

Most modern boiler plants have two valves at the boiler outlet. A non-return valve, placed next the boiler to prevent steam backing up into the boiler when cold or in case of accident to a tube; next to this is placed a straight-way valve with outside screw and yoke. A drain pipe with valve is placed between the two, which is opened before the gate valve is opened and all water allowed to drain out.

The tendency in the past has been to make steam outlets too large, involving an unnecessary expense for valves and fittings. At present steam outlets are calculated on the basis of a steam velocity of 6000 ft. a min., when the boiler is working at rated capacity. Assuming a 500-hp. boiler would require a 6-in. outlet for this rate of flow at capacity, the pressure drop in

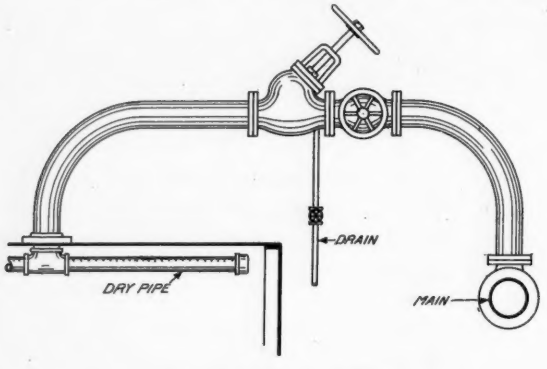


FIG. 3. DRY PIPE AND STEAM CONNECTION

30 ft. of pipe from boiler to main would not exceed ½ lb. If the boiler were worked at 200 per cent rating, the pressure drop would not exceed 2 lb.

DAMPERS

EACH boiler furnace should have a damper that can be fully operated in all positions and operating chains or levers should be brought to the front of the boiler where they can be handled conveniently by the fireman.

Where several boilers are served by one stack, the main damper is sometimes controlled by an automatic damper regulator. This generally consists of a diaphragm operated by the changes in the steam pressure, this diaphragm operating a pilot valve which controls water pressure to a cylinder fitted with a piston which moves the damper. With some types the pilot valve is operated electrically. One type carries the damper wide open until the steam pressure reaches a predetermined pressure then closes the damper entirely; while another type varies the damper opening gradually with the changes in pressure. While the former method probably gives closer regulation, the latter method is usually the better and will operate from open to shut with about 3 lb. variation of pressure.

STEAM JETS

THESE are sometimes used in stationary plants to increase the draft, but can be considered only a temporary makeshift to help on overloads or other emergencies. Several jets would be needed in a stack of any considerable size and should be placed in one or more rings distributed evenly about the cross section of the stack. Steam jets are sometimes arranged to blow into the ash pit to create a forced draft and patented nozzle arrangements for this purpose are on the market.

Size and arrangement of steam jets for draft is largely a matter of experiment in each particular case.

The advantage of this system is its cheapness for installation and repair. Its disadvantage is noise and the great amount of steam required to operate being from 5 to 10 per cent of that generated.

FUSIBLE PLUGS

ALL high-pressure boilers should be protected against low water by a fusible plug filled with pure tin. This plug should be placed in the boiler shell drum or flue (depending on the type of boiler) at the lowest safe water line and where the hot furnace gases will come in contact with it. The plug should be kept cleaned on both the fire and water ends.

When the plug shows signs of leakage around the tin filling, the tin should be melted out, the bore cleaned and tinned then filled with pure tin.

STOKERS AND METHODS OF DRIVING

STOKERS of the chain grate or overfeed type requiring but a small amount of power for their operation are usually driven by small steam engines. With these the rate of coal feeding can be controlled over a wide range either by hand or by an automatic regulator. In the case of the underfeed type of stoker where a forced draft fan or blower is used in addition a variety of drives may be used.

A steam engine having an automatic regulator which causes the engine speed to vary according to the steam pressure can be used to drive the fan and also the stoker mechanism, thus varying the amount of coal fed and also the air supply, according to the demand for steam. This arrangement is largely used in small plants.

An arrangement sometimes used is to provide one large fan for supplying air to several furnaces and having a steam engine on each end of the fan shaft, either engine being large enough to drive the fans, thus keeping one engine always in reserve.

Means are provided for changing quickly from one engine to the other. In some installations small steam turbines can be used to advantage instead of engines.

In larger plants it is sometimes found more economical to drive the fans with constant speed motors and regulate the blast pressure by means of automatic dampers. In this case a separate drive, usually a small engine, is used to operate the stoker or some types of underfeed stokers have a steam cylinder for operating each stoker.

Stoker plants with automatic regulation are in operation where the steam pressure variation is less than 3 lb. during a period of 24 hr.

In the setting of boilers, one of the most important points is proper arrangement and tightness of baffles. Theoretically the gas passages should diminish gradually in area from fire box to breeching in proportion as cool-

ing of the gases decreases their volume. This is difficult to accomplish with the ordinary form of iron flame plates, and firebrick wall construction, but can be secured with inclined baffles built of plastic material and tile. This funnel-shaped arrangement, with the baffle inclined forward from the top of the bridge wall has, also, the advantage of aiding to keep all parts of the gas passages full of hot moving gas, thus avoiding dead corners.

For old boilers, the plastic construction has the added advantage that it can be moulded tight around bent tubes, thus giving a tight baffle when rebuilding, even if the baffle be built upon the original flame plates. To replace a tube, the hole in the baffle can be somewhat enlarged, and, after the new tube is in place, the hole is filled with plastic material again. This construction is, therefore, of essential aid in preventing leakage through baffles, which is a cause of considerable loss. Even if the baffles are of usual fire brick construction, the plastic may be used to advantage in stopping up the crevices.

The harm from leaky baffles is twofold; they permit hot gases to pass direct from firebox to stack without giving up their heat to the boiler; also they permit a high draft in the firebox which draws in excess air, lowering the temperature of the gases, and interfering with regulation of the combustion rate.

To maintain ideal conditions, just sufficient air should be forced through the fire to produce the desired rate of combustion, and just sufficient suction maintained above the fire to carry away the gases as rapidly as they are formed. The former can be accomplished by regulation of pressure under the grate in a closed ashpit; the latter by regulating the damper opening or induced draft suction. Automatic control can be secured by various systems.

Passing all needed air through the fire insures that it meets the combustible at a high temperature; using minimum draft above the fire results in slow passage of gases over the heating surfaces, giving a maximum of time for absorption of heat. Thus the two conditions for effective operation of the heating surfaces are secured, tending to high efficiency and large capacity.

Economizer Practice

SAVING TO BE EXPECTED; CARE AND OPERATION. BY RAINES KESSLER

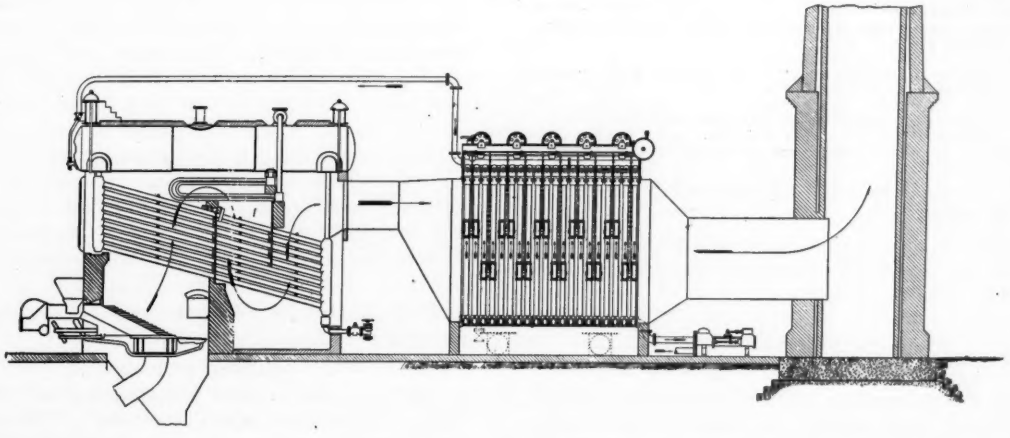
ECONOMIZERS CONSIST of a number of cast-iron tubes and headers through which the feed water passes on its way to the boiler and around which the waste gases of combustion pass on their way to the stack. The purpose of the economizer is to heat the feed water to a high temperature before it enters the boiler by means of the heat contained in the flue gases when they leave the boiler.

Rise in feed-water temperature thus gained from heat which would otherwise be wasted results in a greater evaporation in the boiler per pound of fuel burned and a corresponding fuel saving, or if desired, the same amount of fuel may be used with a proportional increase in the steaming capacity.

AVERAGE SAVING

AVERAGE saving or increased capacity obtained by installing an economizer varies from 10 to 15 per cent, depending upon plant conditions, although in some cases greater savings have been made. Economizers show an unusually high return on the investment, due to the large saving in proportion to the small cost of installation and they should be used in almost every boiler plant, especially as the conservation of fuel is a matter of national importance.

The standard boiler rating is 10 sq. ft. of heating surface per horsepower and is based on the evaporation of 34.5 lb. of water per hour from and at 212 deg. F. by the heat transmitted through that amount of heating



ECONOMIZER ARRANGED ON COUNTER-CURRENT PRINCIPLE. WATER AND GASES FLOW IN OPPOSITE DIRECTIONS

surface or, in other words, the transmission of 33,480 B.t.u. per hour. Actual operating conditions, however, vary greatly from this, as the boiler may be operated at some point below this rating or perhaps at 400 per cent of rating.

A percentage of the heat in the fuel always leaves the boiler in the flue gases. Table I shows the percentage of the heat of combustion present in the flue gases under various conditions. Without the use of an economizer this heat would all be wasted; but by applying the "counter-current" principle, and using the heat in the flue gases after they leave the boiler to heat the feed water before it enters the boiler, a large part of this heat may be saved.

By using an economizer the temperature of the feed water may be raised from the water supply, hot well or exhaust steam feed-water heater temperature to a temperature approaching that of the water in the boiler. The higher feed-water temperature thus obtained, results in a proportionate saving in fuel as the transmission of less heat by the boiler surface combined with the heat recovered by the economizer will produce the same

TABLE I. HEAT OF FUEL PRESENT IN FLUE GASES

Flue Gas Temperature Deg. F.	Pounds air per pound combustible in fuel		
	18 A Per Cent	24 B Per Cent	30 C Per Cent
300			12.4
350		12.0	14.9
400		14.0	17.4
450	12.2	16.1	20.0
500	13.8	18.2	22.6
550	15.4	20.3	25.2
600	17.0	22.4	27.8
650	18.5	24.4	30.4
700	20.1	26.5	
750	21.7		
800	23.2		

A - represents average underfeed stoker operation with forced draft.

B - represents average overfeed or natural draft stoker operation.

C - represents average operation with hand firing and natural draft.

amount of steam as would be produced if an economizer were not used.

The actual saving that any economizer installation is making can be easily calculated from the following formula:

$$\text{Percentage saving} = \frac{100\,(T - t)}{(H + 32) - t}$$

where T = Temperature of feed water after heating.
t = Temperature of feed water before heating.
H = Total heat in steam above water at 32 deg. F.

For example, assuming a feed-water temperature entering the economizer, of 180 deg. F., a final temperature leaving the economizer at 310 deg. F. and a boiler pressure of 150 lb. per sq. in. above atmosphere; the actual saving would be:

$$\frac{100\,(310 - 180)}{1225.5 - 180} = 12.44 \text{ per cent saving.}$$

Results under various conditions of feed-water temperature and boiler pressure may be approximated by estimating 1 per cent saving of fuel for each 10 to 11 deg. F. added to the temperature of feed water.

Knowing the temperature of the gases leaving the boiler and desiring to find the saving that may be made by using an economizer to heat the feed water by means of the heat which would otherwise be wasted, take the temperature of the gases in the boiler uptake and assume a final gas temperature entering the stack of 300 deg. F. in order to obtain the approximate drop in temperature through the economizer. Then, as each 20 deg. F. taken from the gases under average conditions will heat the feed water 10 deg. F., this will give the estimated saving.

The size of economizer necessary to secure these results may be approximated by referring to the dimensions in Table II and allowing 5 sq. ft. of economizer

TABLE II. GENERAL DIMENSIONS OF GREEN'S ECONOMIZERS

No. Tubes wide	Width Inside Chamber Walls			Free Area Through Economizer				Wt. of Section Full of Water	Ext. Htg. Surf. per Section
	For Dampers as Below								
	None	1	2	Length Tube Ft.	Sq.Ft.	Sq.Ft.	Sq.Ft.	Lb.	Sq.Ft.
	Ft.In.	Ft.In.	Ft.In.	9	16.60	23.85	31.10	1636	51.0
4	3 - 4	4 - 1	4 - 10	10	18.44	26.50	34.55	1756	55.8
				11	20.28	29.15	38.00	1877	60.7
				12	22.13	31.80	41.46	2005	65.4
				9	21.85	29.10	36.35	2388	76.5
				10	24.27	32.32	40.38	2570	83.8
6	4 - 8	5 - 5	6 - 2	11	26.70	35.55	44.40	2751	91.0
				12	29.13	38.78	48.46	2942	98.3
				9	27.00	34.25	41.50	3096	102.0
				10	30.00	38.05	46.11	3337	111.7
8	6 - 0	6 - 9	7 - 6	11	33.00	41.86	50.72	3578	121.4
				12	36.00	45.66	55.33	3835	131.0
				9	32.25	39.50	46.75	3760	127.5
				10	35.83	43.88	51.94	4061	139.6
10	7 - 4	8 - 1	8 - 10	11	39.41	48.27	57.14	4363	151.7
				12	43.00	52.66	62.33	4684	163.8
				9	39.25	44.75	51.50	4380	153.0
				10	43.61	49.72	57.22	4742	167.5
12	8 - 8	9 - 6	10 - 3	11	47.97	54.69	62.94	5104	182.0
				12	52.33	59.67	68.67	5468	196.6

Length over Economizer is 2 ft. 5 in. for each 4 sections in length

heating surface per boiler horsepower developed under average conditions.

Care and Operation

After the economizer is installed, great care must be taken to keep the external surfaces of the tubes, which are in contact with the hot gases, as clean as possible in order to prevent soot collecting on them as this insulates them from the heat and materially reduces the rate of heat transmission, thus reducing the saving that might be obtained. In order to do this in the easiest and most convenient way, the manufacturers provide mechanically operated scrapers for this purpose. These scrapers consist of three parts, each part having a beveled cutting edge at the top and bottom which encircles the tube. These cutting edges are both directed upwards and the parts are so supported that these edges are forced against

the tube on the up stroke, thus dislodging the soot which falls between the bottom headers to the soot pit below the economizer and can then be removed at convenient intervals.

Care should be taken to see that the scrapers do not stick, due to any unusual conditions, in which case the chains passing over the pulleys on the gearing frame simply slide over the surface of the pulley. As the scrapers are arranged in sets cleaning four sections of the economizer, the accumulation of soot due to this would render a large percentage of the heating surface inactive, as the ordinary economizer has from 6 to 12 such groups.

Feed-water should not be introduced into the tubes of the economizer at a temperature which condenses the moisture in the flue gases as this will combine with the sulphur in the gas to form sulphurous acid, which will attack the surfaces of the tubes causing external corrosion. Where the initial feed-water temperature is below 90 deg. F. and might cause this condition, the difficulty may be overcome by bypassing a small part of the heated feed water from the discharge of the economizer back to the feed-water line where it enters the economizer and thus tempering the cold water.

Many waters used for boiler feeding purposes contain impurities which are precipitated when the water is heated in the economizer. Because of the slow motion of the water in the tubes, these precipitates fall to the bottom and the economizer should be blown down at frequent intervals in order to remove them. In some cases, the economizer should be washed out two or three times a year to remove all of the sludge which might accumulate.

The most common scale-forming substances are the bicarbonates of lime and magnesia. These are precipitated in the economizer when the water is heated to about 200 to 210 deg. F. because the bicarbonate is reduced to the monocarbonate, part of the carbon dioxide being set free, and the monocarbonate is insoluble. The sulphates of lime which form the harder scale are not thrown down in this manner, but become insoluble in part above 270 deg. F. so that if this temperature is reached in the economizer they will be precipitated. Some of the salts of magnesia are decomposed at high temperatures and will also be precipitated in the economizer.

Air entrained in the boiler feed water or poor circulation, due to irregular feeding, will cause pitting of the internal surfaces of the economizer and inspections should be made annually to determine the possible presence of this trouble in order to prevent this condition rendering the economizer unsafe for operation.

Economizers

RANGES OF WATER AND GAS TEMPERATURES; INFLUENCE ON OTHER EQUIPMENT OF PLANT; DRAFT REQUIREMENTS; INSPECTION. BY OSBORNE MONNETT

TWENTY or twenty-five per cent of the heat generated in the furnaces of an ordinary power plant goes up the stack as a waste product and the recovery of this heat, or as much of it as is practicable, has always furnished an attractive problem for engineers. Attempts to do this have not always met with success, largely because of misapprehension as to the importance of the hot stack gases in the operation of the plant.

In natural draft installations it must be remembered that while the heat of the flue gases is dissipated into the atmosphere, still it is doing an important work in producing the draft required to burn the fuel and that reducing the temperature of the flue gases, either with an economizer or otherwise, reduces the draft and fuel burning capacity of the plant at the same time. The scheme, however, is inherently sound and when the engineering features are recognized, due consideration being given to draft requirements, the economizer is placed on a solid basis as a legitimate piece of power plant apparatus.

Theoretically the amount of heat that can be saved by an economizer is large. In ordinary cases the temperature of the feed water is raised from that of the hot well, 110, to 250 or 300 deg. F., depending upon the extent of the economizer surface, amount of air supplied for combustion, arrangement of gas passes of the boiler, etc. The temperature of the flue gases is correspondingly reduced from 500 or 600 deg. to 250 or 300 deg. Reduction in gas temperature is commonly about twice the increase in temperature of feed water, the saving having a fairly constant relation to increase in temperature. The old rule of 1 per cent saving of fuel for every 10 deg. rise in feed-water temperature will hold good in this connection. Even where a feed-water heater is used to bring the feed water up to a temperature of 200 deg., the economizer will, under proper conditions, justify itself, raising the temperature another 100 deg.

Unquestionably the economizer finds its best expression in large boiler installations with mechanical draft, the boilers being run at high rates of evaporation, where heavy peaks are encountered and sudden swings in the load make the large reservoir of heated water a desirable feature.

In such a plant high steam pressures would prevail, with resultant higher flue gas temperature, since the temperature of the flue gases in the last pass of the boiler must necessarily be as high or higher than the steam temperature within the boiler. This offers a favorable condition for the economizer.

Also the vacuum carried in such a modern plant will be a factor. With vacuums of 28 or 29 in., as employed in steam turbine work and hot well temperatures of only 80 or 90 deg., there may not be enough exhaust to heat the feed water to a proper temperature for the boilers. Under such conditions, utilizing a portion of the hot flue gases for feed-water heating purposes offers a distinct advance in the economy of the plant.

Circulation is on the counter current principle, the coldest water entering at the point of hottest gases, making the heat transfer most efficient. The amount of economizer surface installed is generally about 5 sq. ft. per boiler horsepower, or one-half the amount of heating

surface in the boilers. Heat absorption in B.t.u. per sq. ft. is at ⅓ the rate of absorption per sq. ft. of boiler heating surface, at rating.

It is important not to restrict the draft any more than necessary in installing an economizer; 45 to 50 per cent of the area of the sections is free opening and a gas velocity of 1000 ft. per min. is ordinary practice, the velocity of the entering gases being 3600 ft. per min.

Owing to the large combined area of the tubes the feed water travels very slowly through the sections, a velocity of less than 1 per cent of that in the feed lines being common. For this reason, the economizer offers an excellent settling chamber for mechanically suspended matter. Also the carbonates of lime and magnesia in the feed water are precipitated at the temperatures available. The sludge can be removed with a hose. Periodical blowing down and washing out is necessary, otherwise the apparatus will become inoperative. However, it is not good practice to depend on the economizer for a feed-water purifier. A good rule is to install a feed-water heater where raw water is to be used, taking advantage of the precipitation and settling out of all possible impurities. Where the feed is taken from surface condensing steam turbines this point is not so important.

To get the most out of an economizer the surfaces inside and out must be kept clean. Owing to the apparatus being frequently in inaccessible locations, there is a tendency to neglect it. Periodical inspection and report should be insisted on covering the following points:

1. External surface. Look for accumulations of soot and especially for leaks, as a leak soon sets up corrosion and builds up a deposit of incrustation and scale among the tubes.

2. Internal surface. See that the interior surface is free from accumulations of sediment and that the sludge is soft and can be easily removed with a hose.

3. Safety valves. Free and in good working condition.

4. Blowoff valves. Tight and easily operated.

5. Flange joints. Packing in good condition and not affected by unequal expansion and contraction.

6. Scrapers. Working freely full length of tubes. Frequently soot will accumulate at bottom and cause scrapers to work only part travel.

7. Gearing and reversing mechanism. In good condition and properly lubricated.

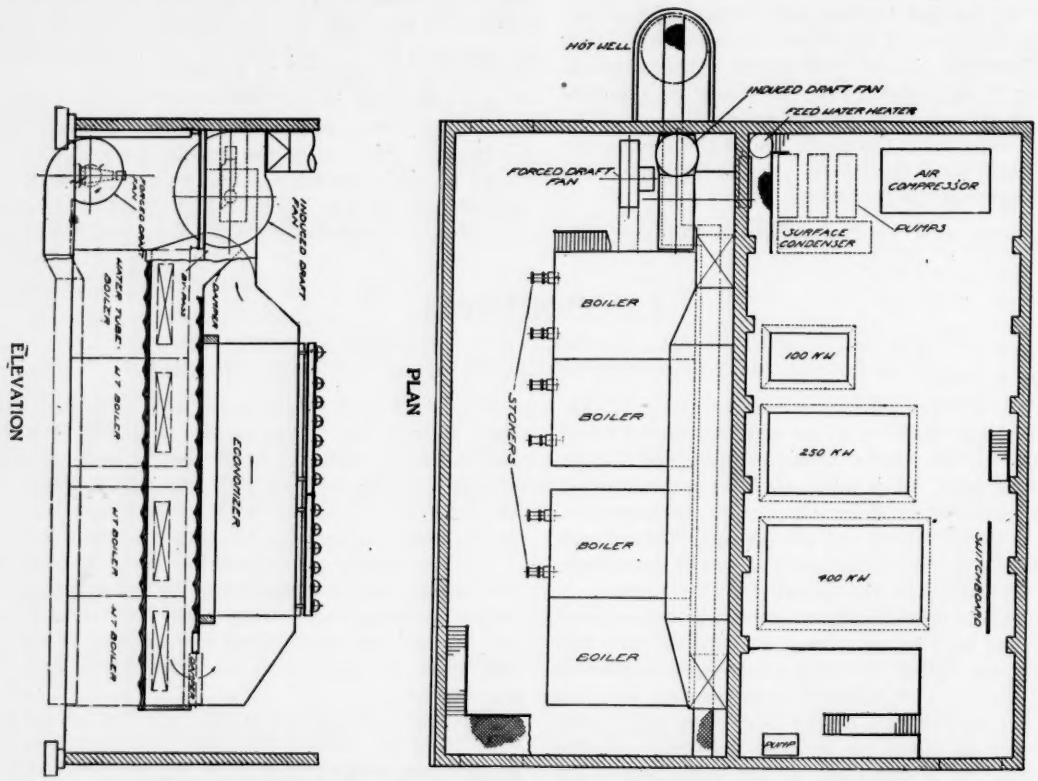
8. Dampers. Working freely and capable of being properly adjusted.

9. Brick work. Free from cracks and air leaks and clean out doors tight.

10. Soot chambers. Properly cleaned of soot and combustion dust.

11. Temperatures. Gases entering economizer. Gases leaving economizer. Feed water entering economizer. Feed water leaving economizer.

Any variation in the normal temperatures, such as an increase in temperature of leaving gases, with corresponding decrease in temperature of feed water, other things being equal, offers a certain check on the condition of the economizer.



TYPICAL ECONOMIZER INSTALLATION WITH FORCED AND INDUCED DRAFT FANS

Troubles of Open Feed-Water Heaters

By J. R. McDermet

THE OPEN feed-water heater, which accomplishes its heating by direct mixing of exhaust steam and boiler feed water, is perhaps one of the most reliable fool-proof devices that enters into power plant operation. It is, however, liable to a few troubles, some of which are matters of proper selection and installation and others purely of operation.

Open feed-water heaters are deservedly liked in power plants because of their ability to handle dirty feed water with continuous high performance. They will handle water of any degree of turbidity with facility, provided it does not contain sticks and stones large enough to clog the water inlet valve. A considerable portion of suspended matter in the water will always collect in the mud zone of the heater, and should be drawn off through the proper blowoff valve on the heater at routine intervals as part of the engineer's duties.

Water inlet valves should preferably be external to permit of proper inspection and maintenance. They are float controlled and operated by the water level in the heater. Sometimes the packing around the float and valve stems becomes so hardened from age as to interfere with float action with the proper pressure to prevent leakage. The remedy is then to renew the packing, and ordinarily it is immaterial what grade of steam packing is used.

Definite rating is given to the modern open-heater by its design, representing a relation between steam volume, water storage volume, and agitating pan area. This relation means something that the manufacturers have worked out through long experience, and should not be ignored in installation, although almost any type of heater will properly heat a quantity of water far in excess of its rating. Over-working cuts down the efficiency of the water storage because a diminished reserve of hot water is available for the boiler feed pumps, and impurities in the water are not given adequate time to separate. It is only where very pure boiler water and a steady load is available that a heater should be overloaded.

Sometimes with waters particularly from artesian wells which contain abnormally high dissolved calcium sulphate or carbon dioxide, the interior of the heater exposed to water will be covered with a coat of scale which may be removed only by mechanical or chemical cleaning. This condition is unusual and does not interfere with the operation of the heater as long as flow through the heater is not restricted. The first difficulty will be with the scaling of the pump suction lines and the boilers, and will serve to draw attention to the condition of the heater.

Contrary to the advertisements of some manufacturers, an open heater is not effective in removing dissolved scale-forming materials although it does have a great advantage in removing suspended matter and dissolved air, either with or without filtering media. It is a matter of common experience with a chemist that all calcium compounds are precipitated from hot solution in a colloidal state and with the passage of time become successively gelatinous and then granular. A colloidal solution may be defined as one having chemically the properties of a mixture and mechanically the properties of a solution. This means, from the mechanical standpoint of the engineer, that impurities will settle out of the water or be removable by filtration only when they become granular and after so long a time that it will not be possible to accomplish the separation in the heater unless a big under-loaded heater with large water storage space is provided. This does not pay. The purity of the water as fed to the boiler will of course improve, due to the presence of condensed steam, which itself is pure.

Oil Trap

Exhaust steam lines in engine service carry a tiny stream of oil, and emulsified oil and water trickling along the bottom. An open heater is provided with some arrangement for trapping out this oil and water and discharging it to the sewer and this device must be kept operative. If oil is allowed to flow into the heater, it will immediately show its presence in the heater gage glass, and when oil shows in the heater it can be suspected also in the boiler. Oil in the boiler combines with scale and produces slugs, which settle on the heating surface and allow it to overheat and blister. Lubricating oil in the boiler is dangerous and prompt decisive measures should be taken to correct the trap on the heater. This is the most serious trouble that can happen with an open heater.

Air Removal

Feed water in passing through an open heater gives up air to the extent of approximately 60 per cent of the dissolved volume. This air when liberated in the interior of the heater reduces the component of the vapor steam pressure and hence the steam temperature, and the temperature of the feed water. A vent connection is provided to exhaust this air and steam into the atmosphere, and it should be piped as directly as possible to the atmosphere without valves or pipe reductions intervening. It is essential also that the vent connection of the manufacturer be utilized since the vent location with respect to the heater is often essential. When the heater is operating under a back pressure, the vent connection should show a continuous trace of steam.

Often, the failure of heaters to deliver water of requisite temperature can be traced to interference with the vent connection. Incidentally a thermometer should always be placed in the pump suction line and desirably a low-pressure steam gage should communicate with the steam space of the heater. As a means of checking one against the other, the rule may be assumed that a heater of standard manufacture by a reputable builder will approach within two or three degrees of the steam temperature corresponding to the steam pressure as taken from steam tables.

Saving Coal by the Use of Feed-Water Heaters

By George H. Gibson

EXCEPTING steam locomotives, few steam plants remain where the boilers are fed with cold water, while exhaust steam is at the same time being wasted to atmosphere. Nevertheless there are many stationary steam boiler plants in which the feed-water heaters are not used to the best advantage. This is sometimes due to the bad condition of the heaters, particularly where closed heaters are used, or it may be due to faulty connections, or to improper heat balancing, which will be taken up more fully in a later paragraph.

Feed-water heaters are of two principal kinds, open heaters in which the steam comes into actual contact with the water, and closed heaters, in which the steam is upon one side of a metal surface, and the water upon the other, and the heat is transmitted from the steam first through the metal and then to the water.

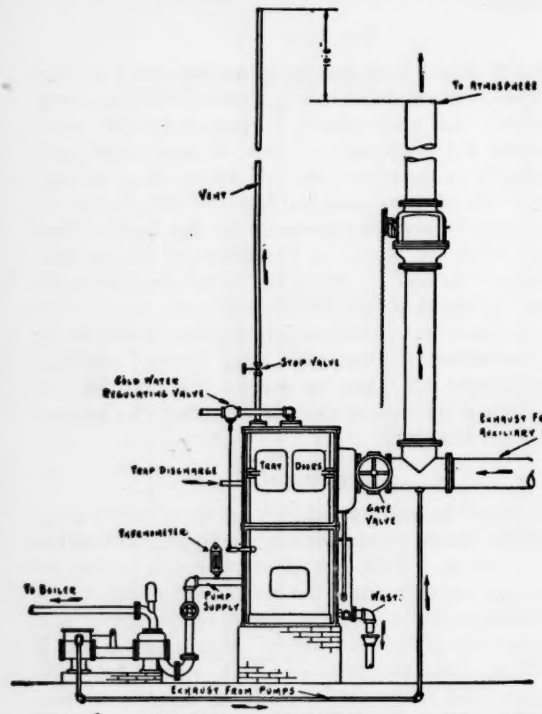


FIG. 1. EXHAUST CONNECTIONS TO OPEN FEED-WATER HEATER. NO BACK-PRESSURE CARRIED

Gain by Heating

To understand just where the feed-water heater comes in the cycle of the steam engine or turbine, let us consider the possibilities of a pound of steam at 150 lb. per square inch gage pressure. Referring to the steam table, its temperature is 366 deg. F. and each pound contains 1177 heat units as compared with water at 50 deg. F. In producing a pound of steam at 150 lb. pressure, 316 units are required to raise the water from 50 deg. to 366 deg. and the remaining 861 units are consumed in evaporation. Now, if instead of feeding the water into the boiler at 50 deg. F., we first raise the temperature to 210 deg. F. by means of exhaust steam, 160 less heat units will need to be imparted to the water and steam while in the boiler, and there will be a saving of $160 \div 1177 = 13.6$ per cent of fuel.

Heat Available in Exhaust

As this heat is to be obtained from exhaust steam, it will be of interest to figure just how much heat there is in the steam after it has passed through an engine or turbine. Let us take as a convenient figure representing a medium-size plant in good condition, 25 lb. of steam per hp.-hr. at 150 lb. pressure as the steam consumption of the engine when running noncondensing. Each pound of steam at 150 lb. pressure represent, as above stated, 1177 heat units added to feed water at 50 deg. F. and the 25 lb. thus represent 29,425 heat units. The exhaust steam leaving the engine contains all the heat not turned into work, except that lost by radiation, and while the latter factor varies according to the type of engine, exposure, etc., let it be assumed at 10 per cent. The heat turned into work can be calculated very nicely, since 1 hp. corresponds to 33,000 ft.-lb. of work per minute, or 1,980,000 ft.-lb. per hour, and 778 ft.-lb. are equivalent to one British thermal unit. A horsepower-hour therefore corresponds to 2,547 heat units, that is, 8.68 per cent of the 29,425 heat units are converted into mechanical power. Adding this to the 10 per cent for radiation and subtracting from 100 per cent we have left 81.32 per cent of the total heat that was in the steam as it left the boiler still available in the exhaust. Even with the most efficient engine or turbine using only 10 lb. per hp.-hr., 75 per cent of the heat is still rejected in the exhaust, the important difference being that the temperature of steam from a noncondensing engine is 212 deg. F., while that from a condensing turbine will be in the neighborhood of 100 deg. F., or lower.

The exhaust from the engine under the above assumptions is not all steam; some of it is water. The part which still exists as steam has the same properties as any other saturated steam at the same pressure, that is, its temperature at sea-level is 212 deg. F. and its total heat above boiler feed water at 50 deg. F. is 1132.32 heat units. Of this amount 970.4 heat units are the latent heat of evaporation, and are given up when the steam is condensed. To raise a pound of water from 50 deg. F. to 212 deg. F. requires 162 heat units. Each pound of exhaust steam in condensing is thus capable of heating $970.4 \div 162 = 6.1$ lb. of water from 50 deg. F.

If the condensed steam itself is added to the water which it has heated, we have from 1 lb. of exhaust steam and 6.1 lb. cold water 7.1 lb. of feed water at 212 deg. In the open feed-water heater this result is attained. In the closed heater the condensed exhaust is usually rejected because it contains cylinder oil, unless an oil separator has been installed in the exhaust connection.

If the amount of exhaust steam available is not sufficient to raise the temperature to 212 deg. F., the engineer

may wish to calculate the temperature that will be reached. For the closed heater this is simple. Suppose, for example, that there is 7 lb. of exhaust steam available for each 10 lb. of feed water to be heated. Let X be the temperature to which the feed water will be raised and t, its initial temperature. Then $10\ (X - t) = 970.4 + (212 - X)$; or if t is 50 deg. F., $X = 153$ deg. F.

For the open heater the problem and formula are a little different. We have, let us assume, 1 lb. of exhaust steam, from which we are to obtain 10 lb. of hot feed water, and as the condensed exhaust is to be added to the cold water, we require only 9 lb. of cold water to start with. Then $9\ (X - t) = 970.4 + (212 - X)$; taking t at 50 deg. as before $X = 163$ deg. F. It will thus be seen that the open heater gives a temperature 10 deg. higher than the closed heater, corresponding to about 1 per cent of the coal bill.

to the temperature to which the feed water is raised. This is because in the closed heater the heat must be transmitted through a solid medium, namely, the tubes, and some difference of temperature is necessary in order to make the heat flow from the steam to the water.

Rate of Heat Flow

The rate at which heat will flow from condensing steam through metal to water depends upon several factors, the least important of which is the resistance of the metal itself, since the conductivity of metal is very high in proportion to the conductivity of water or air. The transfer of heat from the metal to the water is dependent upon the velocity with which the water flows over the metal; the higher the velocity the better the transmission. The presence of any scale deposited from the water will, of course, interfere with the transmission.

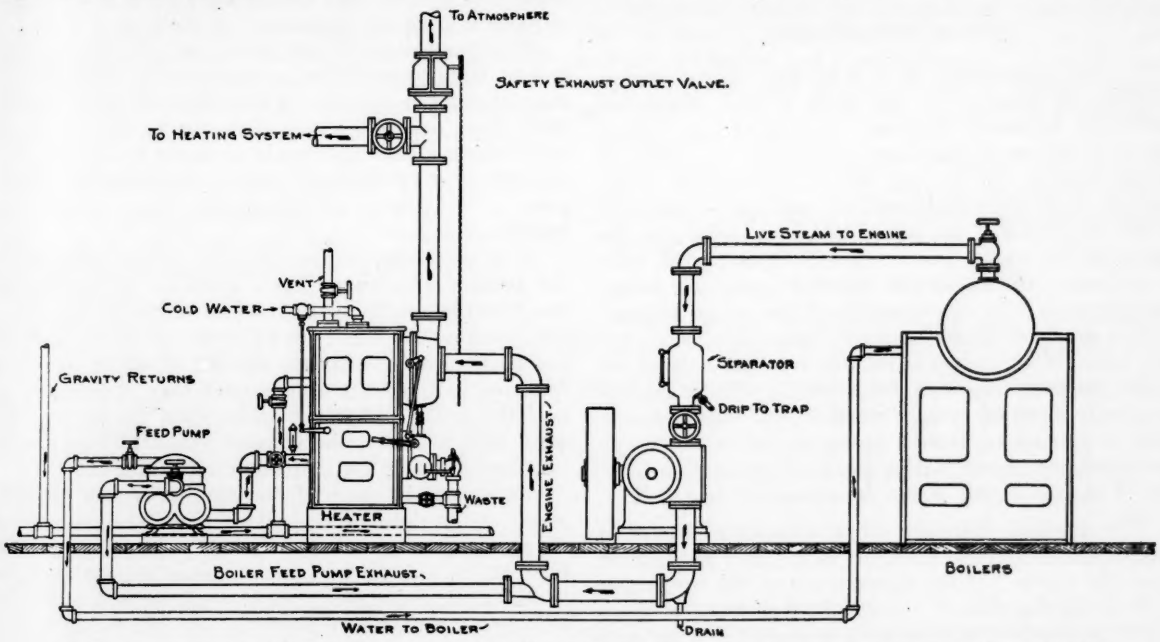


FIG. 2. CONNECTIONS OF STEAM-STACK AND CUT-OUT VALVE HEATER AND RECEIVER TO ENGINE AND EXHAUST HEATING SYSTEM

The foregoing calculations have been made upon simplified theoretical assumptions; for the open feed-water heater they are approximately correct, as such a heater properly connected to prevent waste of steam and vented to prevent the accumulation of air, should heat the water to the full steam temperature, and the only loss will be by radiation from the exterior of the heater, amounting to not over 2 deg. in the temperature of the water. A heater works under these conditions where there is no surplus exhaust or where the surplus exhaust not required by the heater is used in a heating system, low-pressure turbine, etc.

With the closed type of heater, however, we cannot figure upon the water being raised to the temperature of the steam, no matter how much exhaust steam is available. Nor where there is not enough exhaust steam to heat the water 212 deg. can we figure on the exhaust steam and the condensate therefrom being cooled down

On the steam side again the transmission from the condensing steam to metal is very rapid as long as steam only is present; but as soon as there is a trace of air, the transmission is seriously interfered with. Without going further into the physics of this complicated subject, we may say that transmissions in excess of 1000 heat units per square foot of surface per hour per deg. F. difference have been obtained in experimental laboratory apparatus, but that most feed-water heaters and vacuum condensers are designed upon the assumption of a transmission around 300 B.t.u. and this can be depended upon only where there is fairly efficient removal of the air from the heater or condenser. For condensers, special air pumps are provided and in closed heaters a slight exhaust of steam must be permitted in order to carry away the air. The steam circulation through the heater should be so arranged that air-logged spaces or pockets do not form.

Heating Surface

The amount of heating surface required, or the temperature to which the water will be raised by a given amount of heating surface can be calculated from the following formula:

$$S = \frac{W}{R} \log_e \frac{(t - T_2)}{t - T} \text{ where}$$

S = surface in sq. ft.; W = lb. water per hour; R = coefficient of transmission in B.t.u. per sq. ft.; t = temperature of steam; T = initial temperature of water; T_2 = final temperature of water; $\log_e$ denotes the natural or Napierian logarithm.

This formula determines the amount of surface, hence the size of a closed heater.

Size of Open Heater

For an open heater there is no easy, direct way of determining capacity or size, since a large amount of water can be heated to steam temperature in a small space, as witness the familiar injector, the old-fashioned jet condensers, etc. In fact, the size of an open heater depends upon other considerations, and varies according to the conditions of the plant, that is, according to the nature of the water, whether it will deposit much scale in the heater, the amount of storage required for boiler-feed-pump supply, the velocity of the steam through heaters of the thoroughfare type (now little used), etc. The figure of 10 hp. per cubic foot total capacity of the heater has been suggested, but is only a more or less approximate mean of average practice, and should not be used as a criterion, since a heater of less capacity may give much better results than another of greater capacity, but of less intelligent design or improperly installed.

For efficient operation of a feed-water heater, a thermometer should be provided in the feed line leading from the heater. If the temperature of the feed water is below 210 deg. F. and, at the same time, steam is escaping to atmosphere, something is wrong, if it is an open heater. It may be that the heater is air bound, that it is not large enough, that it has not sufficient tray surface for the amount of water being put through it, or that the trays are cemented up with scale so that the water is not properly broken up, but falls at once into the water storage space.

Where an open heater is installed on the "draw the supply" plan, a light back pressure in the exhaust pipe, created by means of a very lightly loaded back-pressure valve, is to be advised. Then if the vent from the heater is kept slightly opened, the heater will free itself of air.

With closed heaters it cannot be expected to obtain water at the full temperature of the exhaust steam, as already explained, but by the formula which has been given, a calculation can be made of the temperature that should be obtained for the given amount of water and heating surface. If actual temperature is below this, caused by scale on the tubes, or air binding of the heater. The remedy for the former is a thorough cleaning and for the latter, forcing the air out, as with open heaters.

Cause of Loss

Probably the most common source of inefficiency in the use of feed-water heaters is the lack of proper heat balancing. The ideal plant would be so proportioned that enough exhaust steam would at all times be available to heat the feed water up to 210 deg. F. and in a condensing plant never more than that amount. That it always pays to have enough exhaust steam to heat the feed water up to 210 deg. F. is apparent from the fact that practically all of the heat not converted into work in the engine, pumps, etc., is returned to the boiler, as against the 80 per cent or so that would otherwise be thrown away in the condenser water, or to the atmosphere. In letting the steam down from boiler pressure to atmospheric pressure, as good use as possible should be made of it; that is, the engines or pumps should be as efficient as possible, and should generate as much power as possible from the steam used in the heater.

If exhaust steam is not otherwise available, it will pay to bleed steam from an intermediate stage of the main turbine or engine. Where there is more exhaust steam from auxiliaries, etc., than can be used for feed-water heating, some use should be found for the surplus, either in heating buildings, heating water for other purposes, or delivery to an intermediate stage of the main turbine or engine.

It is practically impossible so to design a plant that the amount of exhaust steam available from ordinary steam-driven auxiliaries will be just sufficient to heat the feed water to 210 deg. F. at all loads. When the load on the main unit is light, the amount of steam available from the auxiliaries is usually more than is necessary to heat the boiler feed water, while when the load on the main unit is heavy, the exhaust available may not be sufficient to bring the feed water up to 210 deg.

As a remedy some of the auxiliaries can be both motor and turbine driven, which permits of regulating the auxiliary exhaust to the amount required to heat the feed water. If, for any reason, complete motor drive is preferred, the highest economy will be secured by installing an auxiliary generating set exhausting to the feed water heater. The surplus exhaust can be bypassed to the main turbine, during period of light load or the set can include a motor generator connected to the main bus-bars and can have its steam supply controlled by the back pressure in the feed-water heater. It will then take power from the main unit whenever it tends to produce more exhaust than is required for feed heating and will supply current to the main circuit whenever the back pressure and the temperature in the heater tend to fall.

This arrangement is better than the use of independent steam-driven auxiliaries throughout, in that it makes better use of, in other words, produces more power from, the steam which is used for preheating, and it is better than the motor-driven auxiliary plant in that it provides hotter water for the boilers, is thermodynamically more efficient than the main units and is independent of the latter.

To find the safe working load for a Manila rope of given size square the circumference in inches and divide by 7 for the load in tons.

Testing Reciprocating Pumps

INSPECTION TO DISCOVER FAULTS IN ERECTION, LOCATION, ADJUSTMENTS AND ACCESSORIES; TESTS TO INDICATE WHERE LOSSES OCCUR IN OPERATION. BY R. E. TURNER

RECIPROCATING PUMPS in power plants are either of the direct acting steam driven type or power pumps driven by a direct-connected motor on belt. In any case they positively displace the water handled except for the loss due to slip and leakage so that testing for the purpose of securing highest operating economy involves first a thorough inspection and then a trial run during which measurements of water pumped, pressures developed, and steam or power used are carefully taken.

INSPECTION

FIRST OF ALL, a pump must be properly installed for the work it is to perform. It should be attached firmly to a foundation of ample weight to prevent vibration which would tend to loosen pipe joints and cause leakage, and the location should be such that it is readily accessible and at the same time as near to the source of supply and point of delivery as possible to prevent loss of power due to friction in piping.

TABLE I. VARIATION OF SUCTION LIFT WITH WATER TEMPERATURE

Vacuum Ref. to 30" Barom., In.	Abs. Press. in Suc'n. Pipe, Lb. Sq. In.	Calculated Lift, Ft. Hd.	Practical Lift, Ft. Hd.	Max. Temp. of Suc'n. Water, Deg. F.	Practical Temp. of Suction, Deg. F.
1	14.21	1.13	0.8	210	156
2	13.72	2.26	1.6	208	152
3	13.23	3.40	2.4	207	148
4	12.74	4.53	3.2	205	143
5	12.25	5.66	4.0	203	140
6	11.76	6.80	4.8	201	137
7	11.27	7.93	5.6	199	132
8	10.78	9.07	6.4	197	129
9	10.29	10.20	7.2	195	124
10	9.80	11.32	8.0	192	122
11	9.31	12.44	8.8	190	118
12	8.82	13.58	9.6	187	115
13	8.33	14.71	10.4	185	111
14	7.84	15.85	11.2	182	108
15	7.35	16.98	12.0	179	104
16	6.86	18.11	12.8	176	100
17	6.37	19.24	13.6	173	97
18	5.88	20.36	14.4	169	94
19	5.39	21.49	15.2	165	90
20	4.90	22.63	16.0	161	87
21	4.41	23.76	16.8	157	83
22	3.92	24.87	17.6	152	79
23	3.43	26.00	18.4	147	77
24	2.94	27.10	19.2	141	73
25	2.45	28.25	20.0	134	69
26	1.96	29.40	20.8	125	67
27	1.47	30.54	21.6	115	63
28	0.98	31.68	22.4	101	57
29	0.49	32.82	23.2	79	53
30	0.00	33.95	24.1		

Inspection will also include measurement of suction lift to see that this is not too great for the temperature of the water handled, and examination of the piping to see that it is of proper size, to give a velocity of not more than 3 ft. per sec., and is as direct as possible. All water valves should set squarely without leaking and packing should be just tight enough to hold without leaking or causing undue friction.

To test the leakage of an inside plunger it is most satisfactory to block the piston in an intermediate position and remove the cylinder head. Then water at full discharge pressure is admitted to the cylinder behind the plunger. The amount of leakage over a given time can readily be determined by catching the water and weighing it. This method presumes that the valves are tight. Discharge valves may be tested while the head is off by applying full pressure above the valves. To test the discharge valves, pressure is applied within the cylinder; any leakage will then be shown at the suction, or if the head is left off leakage will be shown by disappearance of water covering them. In case it is not practicable to remove the cylinder head to make the plunger test, the leakage may be taken from the opening provided for observation of the suction valves.

Steam driven pumps require close inspection of steam valves to determine the correctness of the setting to secure full stroke without striking the heads and that the valves and pistons are not leaking. Make certain that the sizes of steam and exhaust piping are proper and are as straight and short as possible. Examine the pump governor, also the feed water or pressure regulator which operates it, to be sure that they perform their functions properly.

Inspection of the driving end of power driven pumps should consist in determining the correctness of alinement of pump and motor on belt shafting and condition of driving belt on gear.

SERVICE TESTS

IN THE POWER plant, the principal object of running a test on a pump is to determine where the losses are and how they may be reduced. In order therefore to separate the various losses, the pump is subjected to tests from the results of which the volumetric efficiency, the hydraulic efficiency and the mechanical efficiency are determined.

TABLE II. NECESSARY INLET HEADS FOR WATER OVER 160 DEG. F.

Temp. of Inlet Water Deg. F.	165	170	175	180	185	190	195	200	205	210
Practical Suction Pressure Head Ft.	1.6	2.7	3.8	5	6	7	8	9.2	10.3	11.5

Volumetric efficiency is the ratio between the amount of water actually delivered and the actual displacement volume. The volumetric efficiency is therefore reduced by valve leakage, piston leakage, leaking stuffing boxes, short stroke and slip. It is also affected by speed, leaks in the suction connection and vapor rising from hot water. Low volumetric efficiency therefore calls for an examination into all of these conditions. Volumetric efficiency should not be be less than 90 per cent and may with careful attention to the points mentioned above be increased to 95 per cent or more.

By knowing the hydraulic efficiency an engineer is enabled to determine the losses due to friction of the liquid passing through the passages of the pump. It is the ratio between the total head pumped against and indicated head; i. e., the total head plus the hydraulic losses. Hydraulic losses consist of two general parts; i. e., suction and discharge. Losses on the suction side

of the pump are made up of velocity head, entrance head, friction in suction pipe, losses in the bends, and loss in passing through the suction valves. The discharge losses are made up of the losses due to passing through the discharge valves, the head set up by the valve spring tension, the friction in passing through the remaining passages and the velocity head.

To determine the exact loss due to each minute item requires very accurate and complete tests which are not readily conducted in a power plant and from the standpoint of operation such tests are not necessary as an improvement in some of those losses can only be made by a redesign of the pump. The suction losses, however, are quite readily separated from the discharge losses and this separation should be made in order that an analysis will point the way to a method of reducing the losses.

Referring to Fig. 1, which is a typical water pressure card taken from a direct-acting pump, the method of separating the suction and discharge losses will be noted. This method consists in taking the indicator card in the usual manner then drawing on the card straight lines representing the suction and discharge pressures, the distances from the atmospheric line being proportioned to the scale of the indicator spring used.

In using this method precaution must be taken to see that the gages used in measuring suction and discharge pressures check accurately with the indicator. These readings should be taken while the pump is not working but still holding full suction.

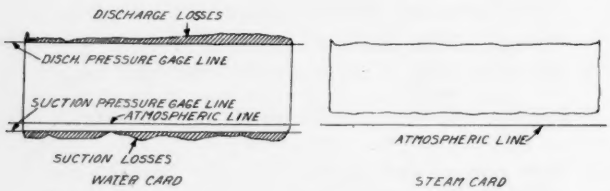


FIG. 1. TYPICAL INDICATOR DIAGRAMS FROM DIRECT ACTING STEAM PUMP

The mechanical efficiency is the ratio between the power represented by the water diagram and that delivered to the pump either from line shaft or motor. Mechanical losses are due to the mechanical friction of pistons in the cylinder, packing boxes and all bearings. They can be determined only by tests, but vary with size, service performed and type of pump. Poor adjustments, misalinements, tight packings, speed, size, etc., all have their influence upon mechanical efficiency.

What may be expected from pumps of several types in good working condition is given in Table III, which is taken from Nickel's book on Direct-Acting Steam Pumps. In this table the mechanical efficiency given includes the hydraulic efficiency, which is quite common practice. In other words, the values given are the mechanical efficiency, as previously defined, times the hydraulic efficiency.

The total efficiency of a pump is the product of volumetric, hydraulic and mechanical efficiencies. In conducting a working test a form should be made out and readings taken, say, every 15 min., during the test in order that fair averages may be secured. This form should include space for all the data needed to determine the above mentioned efficiencies. In testing power plant pumps it is not usual to measure steam consumption except for pumping engines so this will not be considered here. Inasmuch as full boiler pressure is used, the full stroke of the pumps in most direct acting pumps there is practically no initial condensation and with tight valves and pistons a close estimate of the amount of steam used can be made. For ordinary purposes then, the following form will suffice.

1. Date of test.
2. Duration of test.
3. Diameter of steam cylinder.
4. Diameter of piston rod.
5. Diameter of water plunger.
6. Diameter of plunger piston rod.
7. Displacement of plunger (cu. ft.) { Head end / Crank end / Total
8. Average number of strokes per min.

TABLE III. SPEEDS AND MECHANICAL EFFICIENCIES OF PUMPS

Stroke, in.	Normal speeds: Piston, plunger, outside packed: Simple cyl. and compound: R.p.m.	Ft. per min.	Triple expansion: R.p.m.	Ft. per min.	Pressure and wet vacuum pumps: Ft. per min.	Boiler feed: Thick liquids: Ft. per min.	Mechanical efficiencies, per cent. (including hydraulic efficiency): Piston, plunger and ring, 150 lb.	Outside packed, 300 lb.	Press. pumps: 1000 lb. per sq. in.	3000 lb. per sq. in.
3	80	40	.....	....	32	20	50	47	45	39
4	75	50	.....	....	40	25	55	52	50	43
5	72	60	.....	....	48	30	60	57	54	47
6	65	65	.....	....	52	33	65	61	58	51
8	56.2	75	.....	....	60	38	70	66	63	55
10	48	80	64	90	64	40	75	71	67	58
12	45	90	80	100	72	45	77.5	74	70	60
15	40	100	44	110	80	50	80	76	72	62
18	36.7	110	40	120	86	55	82.5	78	74	64
24	30	120	32.5	130	96	60	85	81	77	66
30	25	125	26	140	100	63	85	81	77	66
36	22.5	135	25	150	108	68	87.5	83	79	68
48	18	145	20	160	116	73	90	86	81	70
60	15	150	17	170	120	75	90	86	81	70

9. Temperature of water, deg. F.
10. Steam pressure lb. per sq. in.
11. Delivery pressure.
12. Suction pressure lb. per sq. in. or inches vacuum.
13. Vertical distance between suction gage connection and center of discharge gage, ft.
14. Total head of water, in feet.
15. Weight of water delivered per hr., lb.
16. Plunger displacement cu. ft. per hr.
17. Weight of water by plunger displacement lb. per hr.
18. Volumetric efficiency, per cent.
19. Slip of pump, per cent.
20. Capacity, gallons delivered per 24 hr.
21. Mean effective pressure, steam cylinder, lb. per sq. in.
22. Mean effective pressure, water cylinder, lb. per sq. in.
23. I. hp., steam cylinder.
24. I. hp., water cylinder.
25. Mechanical efficiency per cent.
26. Hydraulic efficiency per cent.

27. Total efficiency per cent.

The methods of securing most of these data need no explanation; comment on some of the items, however, may be of aid.

Delivery pressure, item 11, is measured by a pressure gage attached to the pipe just beyond the air head, it will probably be found to fluctuate with each stroke of the pump, but this can be reduced somewhat by throttling the gage connections. The correct pressure to use is close to the average of the high and low readings.

Suction pressure, item 12, is measured by a gage attached to the suction pipe at its entrance to the pump and when hot water is handled this is usually in pounds per square inches as the water must come to the pump under pressure. When pumping from a lower level, however, a partial vacuum will be formed, and under this condition item 12 is given in inches of mercury.

Item 13 represents the head due to the pump alone, and it has been customary to specify it as the vertical distance from center to center of suction and delivery

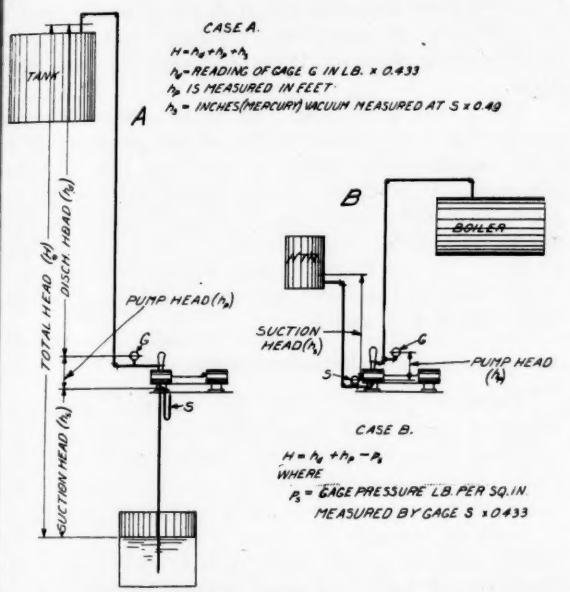


FIG. 2. MEASURING TOTAL HEAD ON PUMP BY MEANS OF PRESSURE GAGES

gages, but accurate tests have proved that the proper measurement is as stated in the form.

Total head of water in feet, item 14, is the vertical distance from the surface of the water in the well to the point, at which it is again exposed to the atmosphere. In power plants much of the pumping work is done in a closed system, water frequently coming to the pump under pressure and being delivered to a boiler or other inclosed tank so that the measurement of total head cannot be made in feet directly. It is then necessary to use gages which give the pressures in pounds per square inch and reduce the results to head of water in feet. Figure 2 shows the two conditions and gives the formulas employed. The constant 0.433 is the pressure due to 1 ft. head of water at 62 deg. F. For water at other temperatures this factor should be changed.

Any one of several methods of getting the weight of water delivered, item 15, may be employed. The most accurate method is to weigh the water in barrels or other containers, but this is not always convenient and requires the constant attention of one man. Any form of feed water meter which has been calibrated may be used or, lacking this, fairly accurate results may be obtained by placing a diaphragm with an orifice in the center between the flanges of a joint and providing pressure gages on each side of the joint. The difference in pressure is a measure of the flow of water and, after calibration, this form of meter will serve for ordinary purposes.

Plunger displacement in cubic feet per hour is found by multiplying item 7, total, by half the number of strokes per hour. And the next item, 17, is found by multiplying item 16 by the weight of water per cubic foot at the temperature at which it is handled.

Volumetric efficiency, item 18, is obtained by dividing item 15 by item 17 and multiplying by 100. Then the slip of the pump in per cent is equal to 100, minus item 18.

The mean effective pressures, items 21 and 22, are figured from the indicator card. In taking cards from the water cylinder the precaution should be observed to use a heavier spring than would be customary with steam pressure of the same value, and it is customary to have two indicators with the connections direct to the cylinder heads.

Items 23 and 24 are calculated by means of the horsepower formula and the mechanical efficiency, item 25, is found by dividing item 24 by item 23.

To find the hydraulic efficiency multiply item 14 by the pressure due to 1 ft. head of water at the temperature at which it is pumped, and divide this quantity by item 22, the result multiplied by 100 will give the hydraulic efficiency in per cent.

As previously stated, the total efficiency is equal to the product of volumetric, mechanical and hydraulic efficiencies.

In the case of power driven pumps, the principles outlined above still hold good; the methods of measuring the power applied vary with the type of drive employed. When belt driven, some form of dynamometer is the most convenient means of measuring the power while the power necessary to drive a direct connected motor driven pump is readily measured by means of a wattmeter.

THE UNITED STATES owns some 5,000,000 acres of oil lands and 53,000,000 acres of coal lands, the title to which belongs to the Federal Government. Since all owners of Liberty Bonds are part owners of this Government, it would be palpably foolish of them to exchange their Liberty Bonds for oil or coal stocks of doubtful values.

Before disposing of his bonds, the holder thereof would do well to consult a banker. Such consultation will not prosper the fake stock concerns, but it certainly will prosper the bond holder.

Tracing Losses in Centrifugal Pumps

WHERE TO LOOK FOR TROUBLE. MAKING TESTS AND PLOTTING RESULTS. BY O. H. DORER

TWO GENERAL types of centrifugal pumps, volute and turbine, are in use. Volute pumps are used for general water supply, circulators for condensers and general cooling water service around the power plant. Turbine pumps are used generally in multi-stage form for high pressure, for boiler feeding, or elevator supply.

Losses in the centrifugal pump may be due to any of following:

1. Operating under wrong conditions of head, capacity or speed with lowered efficiency.
2. Internal pump losses such as leakage.
3. Conditions outside of the pump, particularly the source of supply for the pump.
4. Losses in the driver of the pumps, whether turbine, motor or engine.

WRONG CONDITIONS

RELATIONS existing between head, capacity, efficiency and horsepower are definitely fixed with reference to

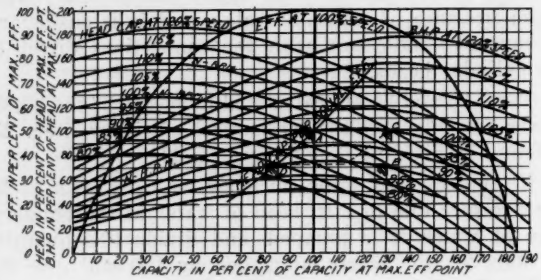

EFF. IN PER CENT OF MAX. EFF.
HEAD IN PER CENT OF HEAD AT MAX. EFF. PT.
BHP IN PER CENT OF BHP AT MAX. EFF. PT.
CAPACITY IN PER CENT OF CAPACITY AT MAX. EFF. POINT

FIG. 1. TYPICAL CHARACTERISTIC CURVES OF CENTRIFUGAL PUMPS WITH SMALL ANGLE IMPELLERS

each other and when plotted in the conventional curves are called characteristic curves. They enable one to understand readily the output of the pump and the importance of operating at the best efficiency. Typical curves are shown in Fig. 1, which gives quantities plotted as percentages of the quantities at the best efficiency.

Point A represents maximum efficiency of pump at N r.p.m. and is the point of head and capacity for which the pump is designed. If the pump is operating at a lower head, as point B where the efficiency is much less, an increase of capacity is obtained over point A. This represents what occurs when the head conditions are incorrectly determined and the pump is working at lowered head.

If operating conditions point to the left of A decreased capacity and increased head would result with a lowered efficiency, which is the common trouble when pumps fail to give the capacity for which they were purchased. Manifestly to keep the efficiency in the maximum when the head is as shown at B, the speed must be reduced until the resultant capacity is as shown at D, along the line of maximum efficiency for the pump in question. This line is readily determined, if we know the values of point A, from the following relation. The heads vary directly as the square of the speed, and at the same time the capacity varies as the first power of the speed. Consequently the horsepower which is a function of head and capacity will vary as the cube of the speed variation. The speed at which to operate the pump to obtain point D is readily obtained from these relations and a curve at a higher speed N_1 is shown.

Now, if it is desirable to obtain increased capacity out of a pump, assuming it is operating at maximum efficiency head as at A, this can be done by increasing the speed to point C, where the efficiency will not decrease as much as for point B. The efficiency for point B is read from the efficiency at N r.p.m. by drawing in the equivalent efficiency line for point C and finding the corresponding efficiency from head capacity curve at N r.p.m.

The power plant man can obtain from the pump manufacturer the characteristic of the pump as all reliable manufacturers obtain a test of each pump before shipment. Observance of conditions under which the pump is actually operating and comparison with these curves will enable the operator to determine the losses under 1. If there is a decided variance between such observations and the shop test, the trouble may be found under losses, 2, 3, and 4.

INTERNAL LOSSES

INTERNAL leakage means that water is being short-circuited through the pump which may occur at any of the fitted joints or between the bushing rings or running joints that are used to reduce to a minimum leakage from the discharge side of the impeller to the suction side.

Leakage through the fitted joints may be due to a gasket being blown out, or metal eaten away, or incorrect matching of parts. Excess leakage through bushing rings is due to wear occasioned by grit in the water, improper material for the service, which under the rotative speed and the eroding action of the water cuts the metal away rapidly. Misalinement and complicated arrangements of bushing rings with temporary seizure and consequent scoring and wear are causes for rapid deterioration of these rings. Operating the pump without water or allowing vapor binding will cause the same difficulties.

Obviously the proper thing to do to prevent loss from internal leakage is to make periodical inspection, replace worn parts and repair fitted joints. Volute pumps being used for medium heads (never in excess of 200 ft.) run in general at lower speeds and, due to construction, are much less liable to internal leakages. Turbine pumps, due to the high heads (from 200 to 1500 ft.) require being made as multi-stage machines, and this makes more internal joints and bushing rings to guard against. Also the construction being along entirely dif-

ferent lines makes the type of joints more difficult. This is especially true in the prevalent split casing machine; therefore, the split-case boiler feed and elevator pump or other high-head service centrifugal should be investigated and inspected more frequently and carefully than the circulating or general service pump. In fact, at least five times a year is not too much to inspect these pumps where in continuous service. Inspection does not mean that repairs are required, but an idea is obtained when the repair will be required to avoid breakdown.

Losses Outside the Pump

Losses in the supply to pumps are such that the pumps cannot operate to full capacity, and are due to improper suction line, air leaks, or insufficient head on the suction in the case of pumps handling hot water.

Where a pump operates under a static suction lift and the losses in friction and velocity head through the pipe occasion a total dynamic suction lift at the center of the impeller, in excess of 18 ft. there will be a reduction in the capacity and efficiency of the machine. For good operation, the total suction lift to the impeller should not exceed 15 ft. Pumps are able to draw water up to a total of 25 ft. suction lift, but the capacity will be reduced 40 to 50 per cent and likewise the efficiency.

Poorly constructed suction lines, especially bell and spigot pipe buried under ground, or lines with a high spot in them are sure to give trouble from air leaks and air binding and cause the pump to lose water or stop pumping altogether. Sharp elbows in the suction of a pump, especially close to the pump inlet cause losses in capacity and efficiency through the eddy currents set up which are carried through to the impeller opening; this element of the machine therefore cannot work properly. Manifestly, an elbow on the suction nozzle of a double suction pump, placed in direction parallel to the shaft, is still worse, as water will be diverted mostly to one side of the impeller, consequently there will be reduction in capacity and efficiency.

The moral of this is to use large suction lines as direct as possible and of flanged piping with long radius ells, the pipe to slope upward toward the pump. Foot valves should be avoided wherever possible on account of the losses in head occasioned by their use.

Where a pump handles hot water as a boiler feeder, head is required on the suction to compress the vapors in the water and prevent the pump from vapor binding. This must be net effective head at the suction of the pump. The amount of head required depends on the design of the pump and too high rotative speed or too much work per stage are determining elements. In general 8 ft. net at the pump nozzle is sufficient for 210 deg. water, but extreme conditions of speed or head per stage may require up to 15 ft. The direct result of too little effective head is loss in capacity for the pump, and vapor binding. Excessively high rotative speeds should be discouraged and plenty of suction head provided for safe, reliable operation of boiler feeders. Venting of casings in case of trouble will help some situations but will fail on others, depending on which element of the above characteristics is mostly at fault.

The use of boiler compounds introduced in the suction side of the pump, should be discouraged on account of the deposit of scale on the interior of the pump. The slight and continuous vibration of the machine breaks off some of this scale which gets churned around the outside walls of the impeller and wears the impeller and stage bushings rapidly. The eroding action of hot water in connection with the higher stage pressures and rotative speed of the boiler feeder is enough cause for wear without the generous help of the scale. For long life in the boiler feeder use good water. Where scale-forming water is used open the pumps frequently (five times a year for continuous operation), clean out the scale and inspect internal parts for wear.

Losses in the Driver

The centrifugal pump generally gets the blame for poor overall efficiency of the unit. Especially are turbine-driven units susceptible to this condition. Frequently the trouble is due to the pump having a slight excess capacity over the designed value and as the auxiliary drive turbines are always bought for so much horsepower with given steam pressures on the throttle and exhaust, there is no overload capacity unless an extra hand operated nozzle is provided on the turbine casing. Consequently, to get the little increased power required on account of the pump being somewhat oversized, these hand operated valves will be opened and the entire machine will lose economy through the turbine nozzles working under partial pressure.

In a case like this the proper course is to allow the normal nozzles to carry the pump at a somewhat lower speed, and in most cases sufficient water is still being delivered. In case of a wide discrepancy, a slight reduction in the impeller diameter will help matters out in view of the fact that the horsepower that an impeller requires varies as the cube of the speed or diameter for a given impeller. The pump manufacturer should be consulted and his advice sought whether or not it is feasible to cut the impeller. The alternative proposition is to increase the size of the main turbine nozzle for the existing conditions. The overload valve of a turbine is of value, but unless the full amount of the overload capacity is absorbed, there is a loss in turbine efficiency.

A further trouble in auxiliary power turbines lies in the fact that while made to give their full output with given steam pressure and back pressure, they are often operated at reduced pressure or different moisture content or increasd back pressure, which items when added together greatly reduce the effective output of the machine and also its economy.

Testing the Pump

Quantities to be determined:

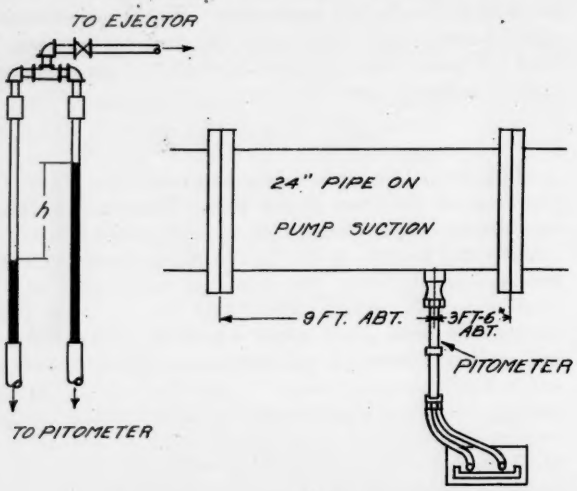
1. Capacity in gals. per min.
2. Total dynamic head at the particular capacity.
3. Speed in revolutions per minute at which the above quantities are obtained.
4. Horsepower input to pump.
5. Mechanical operation of pump.

Determination of the quantities:

1. Capacity

Capacity of a centrifugal pump has commonly been expressed in gallons per minute. The methods of determining the flow may be any of the following: a. Venturi meter or nozzle measurement or good make of water meter. b. Weir—notched or rectangular. c. Pitometer or other flow meter.

(a) A venturi meter with a manometer, when the apparatus is properly installed, so that no swirling eddy



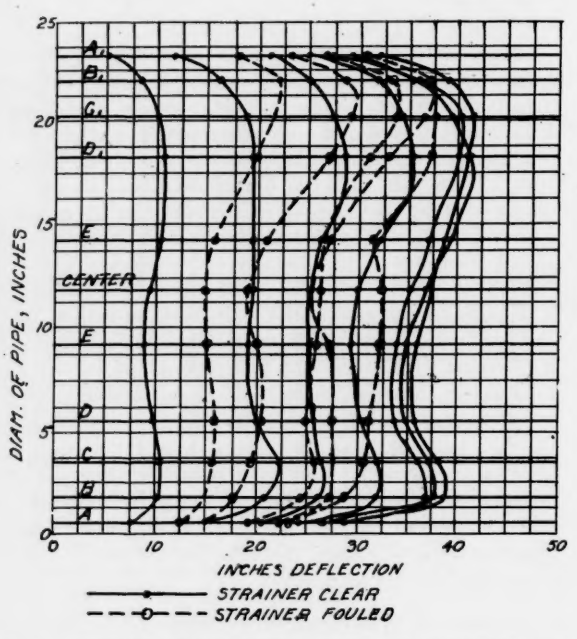


FIG. 2. RESULTS OF PITOMETER TESTS OF CAPACITY AND EFFECT OF EDDY CURRENTS IN PUMP SUCTION

currents can be induced in the flow, gives one of the most accurate and rapid means of determining the flow. The recommendations of the venturi meter manufacturers should be carefully followed when this type of apparatus is installed and a manometer should be used instead of a registering device, as this gives instantaneous readings. The formula for capacity will be readily supplied by the builder of the meter if no manometer is furnished with the meter.

A smooth nozzle such as a fire nozzle can also be used, but it is always best to have some previous calibration of the nozzle by some other reliable means of flow measurement. As most nozzles are used where the pressure available is considerable, there is generally considerable velocity of flow in the nozzle chamber where the pressure reading is taken and the manner in which the pressure gage is placed will affect the result; hence, the necessity of a calibration under conditions identical with those for the test. We would never recommend obtaining the capacity of a nozzle by means of a piezometer, as this always gives large over ratings, and should not be used unless previous calibration is made against another reliable means of getting flow.

(b) The weir, either the rectangular or the notch, is a reliable means of obtaining the flow. The precautions are to see that the proportions are correct and that suitable baffling is used to obtain smooth flow. It is also important to locate the hook gage correctly and with respect to the formula that is being used to calculate flow. When the velocity of approach is considerable this item must be considered also.

(c) The pitometer is a convenient means of getting flow where it is impossible to have any of the other devices, due to physical reasons or other causes. It is most generally used in the suction line of the pump and should always be placed in a long straight run of pipe, if possible. When used in the suction where the pressure is less than atmospheric, some kind of an aspirator should be used to draw the water into the manometer sufficiently to get the deflection or difference in levels. For pitometers placed in the discharge line, air must be pumped into the top of the manometer to get the difference in levels. It is absolutely necessary to integrate the flow by going across the stream taking readings of manometer deflections at definite places in the stream. We recommend to make two traverses as a check at 90 deg. to each other.

The tetrachloride fluid that is usually supplied by the manufacturers of pitometers gives reliable results but the specific gravity of the fluid must be carefully checked for accurate results. Figure 2 shows sample results of a pitometer test, and at same time shows effects of eddy current in suction pipe on the capacity of a pump.

Displacement tests are not recommended except in special cases, as the change of head on the pump while a displacement test is being made gives only approximate results. Where the displacement is a large reservoir the fall or rise in tide must be very accurately determined, otherwise results will be only approximate. Precaution: While capacity readings are being taken, the head and speed should be maintained constant.

2. Total Dynamic Head

For centrifugal pumps this is always specified as feet of water. A pressure gage placed on a pipe under pressure and through which water is flowing at a fair velocity will read the total pressure minus the velocity

head or pressure of flow. Therefore, a gage placed on the suction of a pump reads the static lift plus the velocity and friction head. To obtain the total dynamic head, we must add the suction gage reading to the discharge gage reading and include whatever distance there is between these gages. In case there is a positive head on the suction, the suction gage reading is deducted instead of added. This gives us the total dynamic head where the suction pipe and discharge pipe are of the same diameter. Where the suction pipe is the larger, as is usually the case, we must make correction for the in velocity heads that affect the gage readings. That is, we must calculate the velocity at the suction gage connection and also at the discharge gage connections. The difference in velocity heads to be added to the above total head is the difference in the heads represented by these velocities. Where the discharge pipe is larger, we must deduct this velocity difference head.

3 Speed in Revolutions per Minute

Any reliable speed counter and a respectable watch will do for this.

4. Horsepower Input to Pump

This is rather easy to obtain where the pump is run by a motor, but difficult where other types of prime mover are used.

For motors, all that is necessary is to measure the electrical input and correct for the motor efficiency to obtain the net power to the pump. Most motor manufacturers will state what the efficiency of their motors is under various conditions of load, and this can be checked by core and resistance loss method. But, the full load efficiency can be figured approximately from the name plate rating of the motor. Direct-current motors require a volt and ammeter only. Alternating-current motors should have a power factor meter and an indicating watt-meter in addition.

When the pump is run by a steam turbine, it is well to observe the pressure on the nozzles and the number of nozzles open. No nozzles should be partly open. The steam turbine manufacturer undoubtedly can supply from his shop tests the horsepower-nozzle-pressure curve and corrections can be made, if steam conditions of the test and those of the shop test do not agree. This will give approximately correct results. Where a steam-flow meter is available, that can be relied upon or calibrated by other well-known means, these readings should be observed. Where a condenser is available, the total steam can be measured, but this necessitates running the pump at a constant speed and head during at least 1.5 hr. in order to get reliable results.

Sometimes it is possible to put a dynamometer between pump and driver and determine the horsepower consumption accurately. This presupposes a machine that can be calibrated by some dead weight measurement at a given deflection or dynamometer reading. A dynamometer should always be arranged so that whatever inherent losses are in the machine are not registered on the indicator of the machine. These losses make for additional power required on the part of the driver; and in attempting to check back on the output of the driver correction must be made for them.

Having obtained the quantities, capacity, total dynamic head, brake horsepower, for a given speed, and given working conditions, it is well to obtain a similar set at higher or lower head, maintaining the speed the same. In this way the characteristic curve of performance is obtained.

The pump efficiency or relation of output horsepower to input horsepower can be figured readily. The efficiency can also be stated as a duty in foot-pounds per 1000 lb. of steam in the case of the turbine test.

5. Mechanical Operation of Pump

This item should be investigated for the following:

(a) Do the bearings operate satisfactorily?

(b) If there is a thrust bearing, is this receiving proper lubrication?

(c) If there is no thrust bearing, does the pump shaft have a bad end play, i. e., sudden jerks from one side to the other?

(d) Is there any vibration or noise, and if so, at what head is it noticeable?

(e) Does the suction gage indicate an abnormal lift, indicating restriction or excessive friction?

(f) Is the total head of the pump too great, indicating too much friction in the discharge line for the amount of water passing?

An investigation of these points will lead to the clue to trouble, if there should be any, with the successful operation of the pump.

Help Them Learn English

The English language is not easy to learn, and it is especially difficult for those who come to this country after their school days are over. Therefore, when your foreign associate shows the right spirit by trying to acquire our language, be careful to encourage him in the effort.

If he wants to use a few English phrases, don't rebuff him. Have patience with your foreign-born neighbor, or fellow-worker, when he talks to you in broken English. Never laugh at him or discourage his attempts, but help him to pronounce the words and form his sentences correctly.

In case you speak his foreign language, use English for choice in talking with him, and thus get him into the habit of talking English.

This may seem like a trifle, but it is of vast importance. The strength of a country is its united thought, and we cannot think in common, unless we have a common tongue.

The foreign-born among us will never become 100 per cent Americans until they can talk to us, understand us, read our newspapers and books, listen to our patriotic speakers, and even think in the language of America.

If it seems a waste of time to listen to the attempt of a foreigner to talk English, and if it seems more trouble than it is worth to help him, just consider it as patriotic service and do it cheerfully.

What's in Your Water?

NAMES OF IMPURITIES, OR PLUS, AND THEIR CHARACTERISTICS; TESTING AND TREATING FEED WATER. BY F. F. VATER

WATER IS H_2O nothing more and nothing less. Water is not found in streams, lakes, springs or wells. From these sources it is always H_2O plus, and the plus causes all kinds of trouble in a boiler. It is important that you know what the plus in your water is if you are to rid yourself of trouble. To get all of the facts you should employ a good chemist to make a gravimetric analysis for you. This analysis will tell you of what the plus consists in detail and knowing this, you can learn the cure.

The plus in a water may consist of solids in solution, such as: Silica, Iron Oxide and Alumina, Calcium Carbonate, Calcium Sulphate, Magnesium Carbonate, Magnesium Sulphate, Magnesium Chloride, Sulphate and Carbonate, Sodium Chloride and possibly free acid.

The Silica, Iron and Alumina and Calcium and Magnesium salts are incrusting or scale-forming. The sodium salts being highly soluble, do not form scale, and the only effect they have on a boiler water is to increase its density. Some are of the opinion that sodium chloride is corrosive, but this is not true. This salt is very stable; cannot be separated into its elements at anything less than an incandescent temperature not obtainable inside of boilers.

When corrosion occurs from the use of a water reported to contain it, it is because some chlorine has combined with magnesia or calcium. These salts are unstable and at boiler temperatures will break up into the elements composing them.

All of the incrusting salts named above form scale and increase the density of water as well.

Besides solids in solution there may be solids in suspension which tend to color a water, such as sand, clay, silt, sewage or oil.

The suspended solids do not contribute so much to the scale formation as they do to the accumulation of sludge or mud in a boiler. They do, however, increase the density of a water.

TESTING FOR HARDNESS, ALKALINITY AND TOTAL SOLIDS

IT IS not always convenient to have a careful analysis made of a water. The use of a titrating set will give you all necessary information. A titration of a water is a volumetric analysis and every power plant should be equipped with a cabinet. All determinations (except that of hardness, found by using soap solution) are made by color changes easily recognizable and reasonably accurate. Making a volumetric analysis (or running a titration) you can approximately determine the hardness in your water. No soap solution will precipitate all of the magnesium, and the magnesium sulphate is especially hard to get.

As soap solutions are not accurate where the hardness exceeds 12 or 15 grains per gallon and if you wish to be reasonably accurate, measure out equal quantities of raw and distilled water and mix them. Use the mixture for the determinations.

When diluting with distilled water, you multiply the result you obtain by the factor of dilution to get the actual result.

The soap test gives you the total hardness, i. e., both the permanent and temporary. The sulphates are the permanent and the carbonates the temporary hardness. Next, run an alkalinity test by using methyl orange indicator. This determines the carbonate or temporary hardness. No dilution is desirable in making this test.

If your alkalinity determination is less than the hardness as shown by the soap test, the difference shows the amount of sulphates or permanent hardness the water contains.

Should the alkalinity exceed the hardness, the water contains sodium bicarbonate and you had better take the water to a chemist. Better go to a chemist, because this plus induces foaming, and lap or seam cracks which have caused the explosion of many boilers and it should not be used if it exists in much quantity, unless no other water supply is available.

You can determine its quantity by titrating out to color change with methyl orange indicator and sulphuric acid solution which gives you the alkalinity as before stated. From the result obtained by this titration deduct the determination of hardness by the soap test and the difference will be the sodium bi-carbonate.

The soap test determines the scale-forming matter in a water minus the silica and iron oxide and alumina. These substances, however, are generally present in such small quantity as to be negligible.

To determine the total solids in a water, take a quart and evaporate it to dryness in a water bath. If ebullition is permitted during this operation some of the volatile acids will escape, affecting the result. This evaporation should be done in a porcelain dish. Scrape off the residue and weigh it in grains. Multiply the result by 4 and you have the quantity in grains per gallon.

You can use a less quantity than a quart and less time will be required, but in so doing you increase your error in just the proportion that you reduce the quantity. For instance, if you use a pint instead of a quart your factor will be 8 instead of 4 and your error will be doubled.

From your determination of total solids, deduct the total hardness as shown by the soap test to determine the non-incrusting solids. This is a variation from accuracy to the extent that silica and iron oxide and alumina are present.

A much quicker way to determine the total solids and one which I frequently employ is to use a special hydrometer. You can make it without much expenditure and outside of this use, it should be in every plant to enable you to determine the concentration in the boilers by testing the blow-off water.

Buy a 4-in. seamless copper float for the bulb; have a reinforcing piece brazed to the float. Tap this out to receive the smallest brass bushing you can buy, say 1/4 in.

by ⅛ in. pipe size. Screw a ⅛-in. hollow copper rod into the bushing and solder it. Then opposite the rod cut an opening into the float about 1 in. in diameter and braze to the float a thin piece of copper pipe about 4 in. long, capped on the end. The whole must be perfectly water tight. Place the float in water and put in shot through the opening on top until the float with the rod screwed in permits the rod to stand vertical.

Get a galvanized cylinder 5½ or 6 in. in diameter, 24 in. long; fill it with distilled water, add shot to the bulk until the closed pipe on the bottom of the float is within an inch of the bottom of the cylinder.

Mark on the rod the exact surface of the water. Then weigh ten grains of sodium chloride and dissolve it in the distilled water. Put in the hydrometer and make another mark on the rod at the exact surface of the water.

You will find that if there has been no change in the temperature of the water the addition of ten grains of sodium chloride will raise the hydrometer bulb and hence the rod about ⅛ in.

If you prefer you can add 100 grains of salt to the water and then divide the distance between the first and second marks on the rod into as many equal parts as will plainly show.

In order to use this in determining boiler water concentration, have one mark on the rod with 200 grains of salt in the distilled water, as this is all the concentration allowable in good practice.

This instrument will not be as accurate as an evaporation to dryness in a water bath, but it is sufficiently accurate for all practical purposes.

Tests for Acidity and Oil

To determine the acidity of a water, measure in a Florence flask 250 c.c. and pour it into a beaker. Drop into this water 3 drops of methyl orange. If acid is present the water will turn pink. If it is alkaline, it will turn lemon yellow.

With acid present titrate with caustic soda solution until the pink changes to yellow. The amount of alkali necessary to change the color is equivalent to its acidity.

Oil is sometimes present in water, especially in the condensate returned from heating or drying systems.

Chilling a sample of water is a quick way to learn whether oil is present. Wash the beaker clean; fill it to within a quarter of an inch of the top; pack it in ice and let it stand an hour. If oil exists, prismatic colors will show on the surface of the water if held between you and sunlight.

Effect of Impurities on Boilers

The effect on boilers of the plus in water varies with the character of the plus. Free acid means pitting and corrosion.

Excess sodium carbonate induces foaming and lap cracks leading to tears or rupture.

Suspended matter including sludge and oil, tends to foaming and priming. The calcium and magnesium carbonates produce a scale of large quantity but open and porous and not seriously difficult to remove.

The calcium and magnesium sulphate form a hard dense scale very difficult to remove.

Where the plus in a water is all sulphate, the scale is so hard that two men working steadily will require one day to turbine a 4-in. tube 18 ft. long.

Fortunately, the average water contains both sulphates and carbonates. Where the sulphate content of a water equals 20 per cent of the incrusting solids, the scale will be dense and hard. With less than 20 per cent sulphate not so hard. With 50 per cent sulphate the scale will be very hard.

Another source of serious trouble in boilers is the density of a water. For instance, you are using a treating plant which reduces the incrusting matter in a water to 2½ grains per gallon. You have no trouble with scale; your tubes and heating surfaces are clean, but you continually burn out tubes. The trouble is, excessive density. H_2O is water; its density is one, H_2O plus 200 grains per gallon will give your water a density of 1.0034. If you are evaporating a water containing 15 grains total solids per gallon, 13 hr. of 13 evaporations will bring your density due to concentration, up to 200 grains per gallon, which is the danger point.

This density covers all of the plus a water contains, i. e., the suspended matter, incrusting solids and non-incrusting solids and is the true measure of the value of a water for boiler feed purposes.

The question of the effect the plus in a water has on boilers is always a relative one. There should be an "if" beginning every statement; because the damage done depends wholly on what kind of boilers you are operating and upon the steam pressure carried and the per cent over rating that you work them.

If you have fire tube boilers, carry 80 lb. of steam and are firing an 80-hp. boiler to evaporate 40 hp. in steam, you need not worry much about anything but free acid.

Again, if you carry 100 lb. of steam and work your boilers to 125 per cent of rating, the same water that caused you no trouble at 80 lb. pressure and 50 per cent of rating will bother you seriously.

The higher the pressure carried and the greater the overload factor the more trouble you will have. At 240 lb. pressure, 250 per cent of rating, you will put as nearly H_2O without plus into your boiler as you can obtain if you wish to avoid trouble.

Estimating Treatment Needed

Knowing what the plus in your water is by using the titrating set, you have a basis on which you can closely estimate the quantity of cure required. In the case of 80 lb. pressure, running a boiler at half its rated capacity, a softening plant is scarcely necessary. You can prevent scale formation by preparing your own treatment.

The active base of any and all compounds is soda. It may be in the form of soda ash (sodium carbonate), or caustic soda (sodium hydrate).

Provided the calcium and magnesium salts are all in the sulphate form, soda ash is the cure. For the best result feed into your water regularly one grain of soda ash for each grain of sulphate the water contains. Bear

in mind that the sulphates are the permanent hardness and are found by deducting the alkalinity from the soap solution tests.

Should the water contain an equal amount of sulphates and carbonates, use caustic soda, one grain for each grain of carbonate or temporary hardness your titrations show.

In feeding caustic soda into this water, you have two chemical reactions. First, the caustic soda (sodium hydrate) absorbs the excess carbonic acid changing the bi-carbonate into the mono-carbonate which being insoluble, precipitates. The carbonic acid combines with the sodium hydrate forming sodium carbonate or soda ash which reacts on the sulphates. The sulphuric acid combines with the soda changing the soda ash to sodium sulphate. This sodium sulphate is therefore an end product of the reaction and being highly soluble, remains in the water. It does not form scale and is inert except that it increases the density.

The carbonic acid released by the soda ash combines with the lime and magnesia forming the insoluble mono-carbonates. No compound can do much good when used in a water containing temporary or carbonate hardness only.

The sulphates are the cement or binder that makes scale hard. Their removal therefore, will change a scale from hard and dense to light open and porous.

Should your water contain three grains of sulphate and 12 grains of carbonate, feed six grains per gallon of caustic soda. This will cause the precipitation of the sulphate, and three grains of the carbonates leaving six grains untreated. Should you feed 12 grains per gallon of caustic soda in the water you will remove the sulphates but you will have formed 9 grains per gallon of soda ash in excess and this is too much. The limit of excess soda ash in any water should never exceed three grains per gallon.

Where your boilers are operated at 100 lb. or more pressure and 25 per cent or more overload, compounds should never be used. You have reached the points where water softening pays very attractive dividends.

Examination of Scale to Determine Impurity

This method is out of date. Find what the plus in your water is and you will necessarily know what the scale is. If you are so situated that a number of samples of scale can be examined and you know what the water forming each sample contained, you can get an approximate idea of what the water contains from the appearance of the scale.

Methods of Treating Water

Condense all or practically all of the steam used with a surface condenser and you will have no scale trouble. If your make-up water is very hard, treat it by softening.

Distillation will prevent all scale troubles. Softening water will prevent all scale formation.

As to which of the three methods is best is a question that can only be answered by a careful analysis of the conditions that must be met. It is an engineering problem.

As a rule, distillation costs are prohibitive unless a recovery of the heat is possible.

There are in use four softening processes. Three are by use of lime and soda ash. They are, the hot continuous process, the cold, intermittent process, and the cold continuous process. Then there is the Zeolite process, which can be regenerated by the use of salt. The latter process is successful when treating comparatively good waters, or hard waters. The cold lime and soda ash process is used as a pre-treatment, then this treated water is passed through the Zeolite. Where the lime and soda ash pre-treatment followed by a Zeolite process is used, the installation cost is very high, as is also the chemical and labor cost of treatment.

The lime and soda ash process is in much more general use than other processes.

The value of any process of treating water is measured by three standards: Will the softened water prevent any scale formation? Obtaining this, the other questions are: What will be your chemical cost? What will be your labor cost?

The chemical cost as far as the direct chemical reaction is concerned, is the same in any process, but in any process there is a chemical waste due to the necessity of using an excess of chemical. This chemical waste is measured by the condition of the water after treatment as determined by its causticity and alkalinity.

If a treating process will deliver you a water not harder than 1½ gr. per gallon, with no causticity, and an alkalinity of 1½, there would be no chemical waste. This, however, is not physically possible, but the nearer you can obtain this result the less your chemical cost will be. Again, if the manufacturer guarantees the hardness only of the treated water, he can meet the guarantee by over-treatment and can compel you to stand a waste of ten cents a thousand gallons. This means on a 1000-hp. plant, 24 hr. a day, 300 days a year, a waste of $2880.

By requiring that not only hardness but causticity and alkalinity be guaranteed, you close the door to waste.

The labor cost of operating a softening system varies from ¼ of a cent to 5 cents per thousand gallons, depending on the type of apparatus. It is therefore very important that you ascertain what your labor cost of operation will be. A waste of 10 cents per thousand gallons in chemical and 4¾ cents per thousand gallons in labor will make a softener cost you more to operate it than it will save and there are dozens of cases where this result has been met, and the softeners are standing idle.

Frequency of Sampling and Testing

This is a question of the condition under which you are operating. In a cold, intermittent process you will need to sample every tank full of raw and treated water and the number of samples taken will depend on how many tanks you have and how quickly you empty a tank.

In the continuous process the frequency will depend

on how variable the quality of water is and upon how automatically the feed of chemical cares for load changes.

On streams varying in quality from 7 grains alkalinity to 12 grains free, sulphuric acid, using the continuous process, one sample a day of raw water cares for the variation. Again, in the continuous process one sample and test of the treated water per day of 24 hr. is all that is necessary.

Titration Tests

In the following directions for tests by titration, c.c. is the abbreviation for cubic centimeters; 1 gram = 15.43 grains, and 1 oz. avoirdupois = 28.35 grains.

Measurements are given in the metric system, as that is the one commonly used by chemists, but the indications by color or precipitate are the objects sought, so that the units used for measurement are not of consequence so long as correct proportions are preserved. The method is taken from Publicity Magazine of the Underfeed Stoker Co. of America.

To make the tests requires one measuring tube graduated in cubic centimeters. One glass or enameled ware pitcher with a capacity of one litre, and this must be graduated in grams, cubic centimeters and litres. You will also need apparatus as shown in accompanying sketches or apparatus so that you can do this work.

All of this material can be purchased through any drug store.

We suggest that you buy your chemicals measured out for you and done up in separate packages, each plainly labeled. These you can also purchase from your local druggist.

Oil in Water

When oil in boiler feed water condenses there is considerable danger of damage.

Take 250-c.c. of the boiler feed water, put it in a glass stopper bottle, put in 50-c.c. of pure ether, placing stopper in bottle and shake contents thoroughly for several minutes, then allow bottle to stand 15 minutes, during which time (ether absorbs oil from water when bottle is shaken) a layer of oil-ether solution forms on the top of the water as shown in Fig. 1. Next, get a U-shaped glass tube with one arm longer than the other and syphon water out from under oil-ether solution as shown. Now take oil-ether solution that remains in the bottle and put in a porcelain dish. Place the dish in a lukewarm water bath as shown in Fig. 2. The ether will evaporate, and residue that is left in dish will represent the oil. If dish was carefully weighed when empty, and then with oil residue in it, the difference in weight would represent the quantity of oil contained in 250-c.c. of boiler feed water. To determine the amount of oil in a gallon of boiler feed water, multiply by 15.14.

Hardness of Water

Hardness of boiler feed water has a direct bearing on the scale forming tendencies of the water.

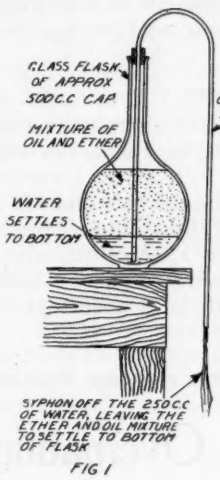
To test for hardness:

First: Make a soap solution by dissolving 13 grams of sodium oleate in 500-c.c. of alcohol and 500-c.c. water.

Second: Make a standard quantity of hard water by dissolving 1.11 grams of pure calcium chloride in one litre of pure distilled water.

Third: Take 58-c.c. of pure distilled water and add to this 12-c.c. of standard hard water, then add 1-c.c. at a time of the soap solution, from a measuring tube, and after each addition shake the bottle thoroughly. The procedure should be repeated until a lather is secured that will stay intact for 5 min. Thirteen c.c. of the soap solution should be required to make such a lather. If less than 13-c.c. of the soap solution is required, this indicates that the soap solution is too strong and it should be diluted with pure distilled water until the point is reached where 13-c.c. of the soap solution will make a permanent lather.

To make the test for hardness of water, rinse out a bottle which is to be used in the test several times with boiler feed water, then put 58.3-c.c. of boiler feed water (which should be filtered if there is much suspended matter in the water) in the bottle and add 1-c.c. of the soap solution at a time, and after each addition shake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WATER SETTLES TO BOTTOM
GLASS SYPHON
SYPHON OFF THE 250 CC OF WATER, LEAVING THE ETHER AND OIL MIXTURE TO SETTLE TO BOTTOM OF FLASK
FIG 1

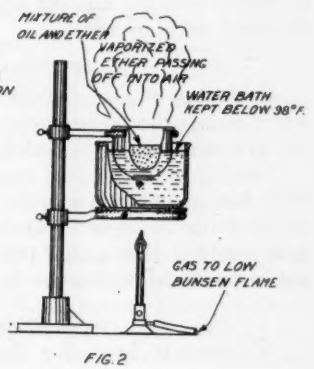


FIG. 1. TEST FOR OIL IN WATER
FIG. 2. EVAPORATION OF ETHER FOR OIL TEST

the bottle thoroughly. Repeat this operation until a lather is obtained which will stay intact for 5 min.

For example, if it requires 14-c.c. of soap solution to produce the permanent lather, the total hardness of your water is equivalent to 14 grains per gallon, or 15-c.c. of soap solution means 15 grains per gallon, etc.

Acids and Alkalies in Water

It is important to know whether your boiler feed water is acid or alkaline, because the acids cause corrosion, and alkalies cause priming.

Test for Alkalies

First: Dissolve one gram of methyl-orange powder in 1000-c.c. of pure distilled water.

Second: Dissolve 40-c.c. of hydrochloric acid in 1000-c.c. of pure distilled water.

Put 200-c.c. of boiler feed water in a white porcelain dish and to this add two drops of the above methyl-orange solution, and the water will then turn yellow.

Next, add the hydrochloric acid solution drop by drop, from a measuring tube, but constantly stir the sample until the color changes to orange and then finally to a faint pink. The number of c.c.'s of acid used gives the number of parts of calcium carbonate per hundred thousand parts of water.

Test for Acids

First: Take 100-c.c. of 95 per cent alcohol and dissolve in it 0.5 gram of phenolphthalein crystals, then add a caustic soda until the solution has the faintest pink color.

Second: Dissolve 21.5 grams of pure carbonate of soda in 250-c.c. of hot distilled water, then add to this solution distilled water until you have 1000-c.c. of this solution.

Put 200-c.c. of boiler feed water in a porcelain dish and to this add two drops of the above phenolphthalein solution. Next, add the carbonate soda solution drop by drop from a measuring tube until a deep pink color appears and stays permanently. The number of c.c.'s of carbonate of soda solution just used should be multiplied by 9.58 to give the number of parts of acid per hundred thousand parts of water.

Sulphate of Lime

Sulphate of lime (calcium sulphate) in boiler feed water causes the forming of a very hard tenacious and fine grained scale in the boilers.

Put 2 in. of boiler feed water in a test tube and add a little barium chloride solution. If the white precipitate which forms remains in suspension after a little nitric acid is added, it is a sign that calcium sulphate is present in the boiler feed water in fairly large quantities.

Magnesium

Magnesium in boiler feed water causes foaming (which makes the liberation of steam bubbles difficult) in the boilers and also often carries with it chloride radicle, which causes corrosion.

Put about 3 in. of boiler feed water in a test tube and bring it to a boil over a gas flame. Next, add a little carbonate of ammonia and a bit of potassium of soda, then if the white precipitate forms after the solution has been allowed to stand a few minutes, it is a sign that magnesium is present in your boiler feed water.

Chlorides

Chlorides in boiler feed water cause corrosion and priming.

First: Dissolve 4.79 grams of silver nitrate in a litre of distilled water and place in a dark bottle.

Second: Dissolve 10 grains of patassium chromate in 100-c.c. of water.

Now, put 58.3-c.c. of boiler feed water in a bottle that will hold about 100-c.c. Next add a few drops of the potassium chromate solution. Pour in the silver nitrate solution slowly, shaking the solution gently until a permanent red rust color is obtained. The number of c.c.'s of silver nitrate solution that you add will indicate approximately the number of grains per gallon of the chlorides present in the water.

Organic Matter

This causes priming and corrosion.

Put a few drops of a strong solution of potassium permanganate in 500-c.c. of boiler feed water. If the red color of the potassium permanganate fades away it leaves the boiler feed water colorless and a large quantity of organic matter is present. If a yellowish tinge remains in the boiler feed water, it is a sign that organic matter is present, but not in large quantities.

Overhauling the Piping

Inspection for Bad Layout, Leaks and Proper Insulation. Checking Up Sizes. Draining and Traps. Testing and Estimating Lossees. By Carl L. Svensen

OVERHAULING OF PIPING should properly include a study of its purpose, the methods adopted for accomplishing this purpose, and the inspection and testing of all piping, valves, and other parts of the system to see how well the purpose is being carried out. Piping is always a part of the power plant equipment and may be the source of many troubles as well as being both necessary and convenient. The causes of trouble arise from improper design, improper installation and materials, additions to and changes from the original design, and improper attention after installation.

Piping in common with all parts of the plant which are in constant use requires constant attention if it is to be kept in a state of efficiency.

The purpose of the piping is to convey steam or water in sufficient quantity and proper quality to the different machines, and to care for the steam and water after use. This purpose should be accomplished with the smallest loss from condensation, small pressure loss, and with safety.

Inspection

The first step in overhauling the piping would be to make a careful inspection of the whole system, making a record of all points noted. A diagram drawing of the piping systems will be found very useful, as a record and to determine the best arrangement for future changes and additions. Such a drawing can be easily made using a simple single line representation. The position and kind of all valves should be noted as well as the size and use of each pipe line, Fig. 1.

The inspection should include a careful search for leaks and the observance of all pipes to see whether they are properly covered or insulated to prevent radiation losses. Note also the thickness and condition of the insulating material. The location of all valves should be studied to note whether they are necessary and are adapted for the purpose intended. See if there are sufficient valves to cut out lines which are unused or occasionally used. Such lines, especially if long should have two valves, one near the main line and one near

the engine or turbine. Extra valves are necessary to cut out a line in case of accident or to change the regular system of operation to prevent shut-downs under emergency conditions.

Leakage Losses

Loss due to a small leak can be appreciated when it is realized that a hole ⅛ in. diameter will allow about 2500 lb. of steam at 150 lb. pressure to pass through in 24 hr. Several leaks can easily make themselves felt at the coal pipe as well as cause damage to insulation, to valves, and in other ways. For this reason all leaks

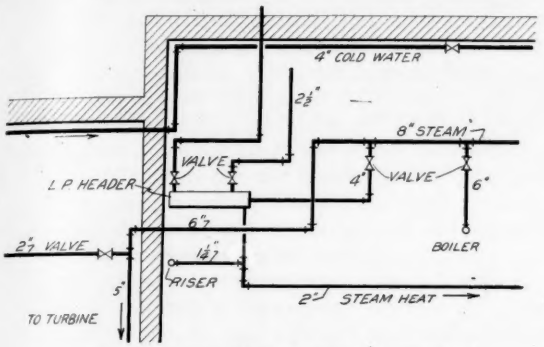


FIG. 1. SINGLE LINE PIPING DRAWING

should be stopped as soon as detected. Leaky joints may be due to a number of causes, the remedy for each of which will be suggested. The principal causes are: Joints too far from supports; lack of provision for expansion; improper supports; insufficient supports; lack of proper alinement before "making up;" excessive vibration; insufficient number of bolts for flanges; poor threads or insufficient number of threads; packing too thick; worn or poor packing for flanges; scored valve stems; insufficient packing or worn valve stem packing; light weight valves; leaks at screw threads with uncovered pipe due to range in temperature. Fittings used to allow for changes in alinement develop leaks, while bends made of a single piece of pipe do not have this fault.

Leaks at joints between pipe and flange can be stopped by applying a pipe-joint clamp, Fig. 2. Split

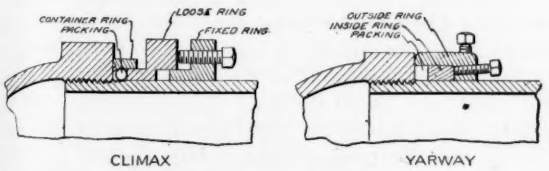


FIG. 2. PIPE-JOINT CLAMPS

seams or rust holes can be repaired by using a piece of sheet packing and an emergency pipe clamp, Fig. 3.

Leaky Valves

Worn valve seats should be repaired as soon as discovered, as it requires but little time for them to be damaged beyond repair. When the weight of piping is carried through valves it will cause them to leak, hence supports should be provided near all valves. Valves should be tested for tightness and if leaky should be put in good condition at once. Sometimes light weight valves will be found, which have had their shape affected by the strains of "making up." Such valves will never be satisfactory and should be replaced by valves having sufficient material to withstand the wear and tear of erection, use and repair.

New pipe lines, and old ones after the placing of new valves or making changes, should be thoroughly blown out to remove any dirt, scale or other foreign matter. Then before closing any valves they should be examined, whenever possible, to see that seats are clean. There should be an evident means of being sure that blowoff valves and safety valves are not leaking, so that such a source of loss can be stopped at once. Small leaks amount to big losses when they occur over a long

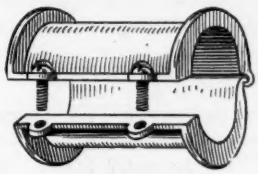


FIG. 3. EMERGENCY PIPE CLAMP

period of time and especially when there are a number of such leaks. Visible leaks are usually attended to at once, but leaks inside of valves are not seen, so that valves require constant attention. For this reason unnecessary valves should not be used.

Cover the Piping

A kind of leak which is not so evident as escaping steam is the uncovered steam pipe. In going over the pipe coverings they should be examined for the presence of moisture, for cracks, shrinkage spaces, looseness from the pipes and fittings, and opening of joints. Wet coverings and openings which allow air to reach the pipes do not prevent loss of heat by radiation. For these rea-

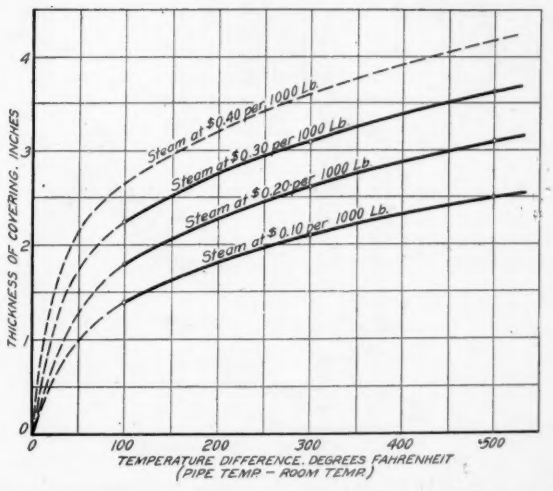


FIG. 4. CHART FOR DETERMINING PROPER THICKNESS OF 85 PER CENT MAGNESIA FOR MAXIMUM NET SAVINGS AT VARIOUS TEMPERATURES AND PRICES OF STEAM

sons all coverings should be thoroughly inspected at least once a year and refitted to the pipe, renewed where necessary, and otherwise put into good repair.

Valves, flanges and fittings should be covered with the same care as is the piping, for the surfaces thus ex-

posed are often a large per cent of the total radiating surface. Provision for getting at the valves and for inspecting the tightness of joints can be made by using molded sections of covering. Bare pipes are not only wasteful of heat but are the cause of moisture in the steam which may lead to dangerous conditions. The loss of heat due to radiation from bare pipes is about 3 B.t.u. per square foot per hour per degree F. difference in temperature between steam and outside air. From 80 to 90 per cent of the loss can be prevented by proper insulation of the pipe.

The thickness of covering to use on piping with high pressure steam varies from 1 in. on less than 2 in. diameter to 3 in. on sizes over 8 in. diameter. The following thicknesses of Nonpareil High Pressure covering are considered economical under average conditions inside of buildings for high-pressure steam.

THICKNESS FOR PIPE COVERING

Cost of Steam per 1000 Lb	Saturated Steam	Superheated Steam
Less than 10 cents	Standard thickness	1½ in. thick
10 cents to 15 cents	Standard thickness	2 in. thick
15 cents to 20 cents	1½ in. thick	Double layer 1½ in.
20 cents and over	2 in. thick	Double layer 1½ in.

Standard thickness varies from 1 in to 1½ in.

Figure 4 is from a very complete set of tests made by L. B. McMillan and reported in A. S. M. E. Transactions, Vol. 37.

PRESSURE DROP

ALLOWABLE drop in pressure varies with conditions. In some large turbine plants with high superheat a

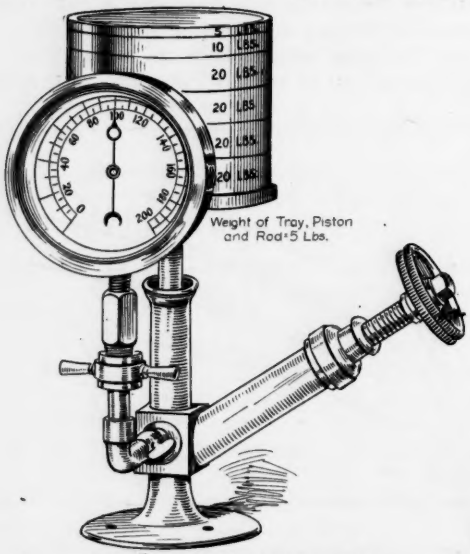


FIG. 5. CROSBY GAGE TESTER

drop of 8 to 10 lb. is allowed. Under ordinary conditions a drop of more than 4 or 5 lb. is excessive and means a large loss of energy.

The drop in pressure is found by reading the gages at the boiler and at the steam engine or turbine at the same time. The difference is the drop in pressure. Steam gages used for finding the pressures should be tested to make sure of correct readings. Pressure gage testers are made for this purpose, one form being shown in Fig. 5. The weights rest upon a movable piston which transmits pressure to the gage through oil contained in the body of the apparatus. Full directions for testing gages are furnished by the manufacturers and need not be repeated here. Readings on the gage should be taken as the weights are increased and also as the weights are taken off. A test record is shown in Fig. 6 which indicates that the gage reads about one pound too low both with rising and falling pressures. The gage hand should be reset.

Pressure drop may be due to a number of causes, such as too small pipe, obstructions in the pipe, too many turns and valves, bare pipes or inefficient covering. For high pressures and superheat, small pipes and high velocities are used with consequent drop in pressure which is provided for by a higher initial pressure. In such cases the first cost is less, radiation losses are less, repairs cost less, and provision for expansion is easier. The pressure drop helps maintain the superheat. Drop in pressure can be remedied to some extent by using valves with ample passageway, by providing a receiver or reservoir near the steam user, by careful attention to pipe covering, by reducing the changes in direction of the piping, and by enlarging the pipe size. Anything which interferes with the flow of the steam will cause a drop in pressure which means a constant loss. This does not mean that boiler pressure should be maintained at the throttle as the large pipes needed would have a larger radiating surface and offset the gain in pressure.

Gage #465668 Make ——— Date ———

Standard Weights	Gage Up	Standard Weights	Gage Down
75	74.5	105	104
80	79.5	100	99
85	84	95	94
90	89	90	89
95	94	85	84
100	99	80	79
105	104	76	74.5

FIG. 6. RECORD OF GAGE TEST

A velocity of about a mile a minute was formerly used as a standard for steam flow in pipes. For superheated steam the absence of water makes the use of much higher velocities possible as the steam flows more freely and radiation losses are less. In large steam turbine plants, velocities up to 8000 and 10,000 ft. per min. are in use. For steam engines, larger pipes are often used to provide a reservoir to meet the varying demand for steam. Smaller piping, with a receiver near the throttle is a better arrangement as a continuous flow of steam at a high velocity can be maintained from boiler to receiver. The receiver must be carefully drained so that dry steam will be supplied to the engine. A large pipe is used between receiver and engine. The use of a receiver separator will remedy the vibration of pipes when due to a changing velocity of steam flow.

The velocity of flow can be figured from the quantity used by the steam consumer or a steam meter may be used such as the St. John, Sargent, Bailey, General

Electric, etc. Full descriptions may be had by writing to the various companies making them.

The usual formula for figuring flow of steam where the pressure drop is small is Babcock's

$$W=87\sqrt{\left(\frac{PDd^5}{L(1+\frac{3.6}{d})}\right)}$$

in which W = weight of steam flowing in pounds per minute; P = drop in pressure in pounds per sq. in.; D = mean density, pounds per cubic foot; d = inside diameter of pipe in inches; L = length of pipe in feet.

Draining Piping

Wet steam is both dangerous and damaging. Water coming over with steam will produce water hammer and cause breakage of parts of the steam using machines. With small steam turbines, vibration and slow down of

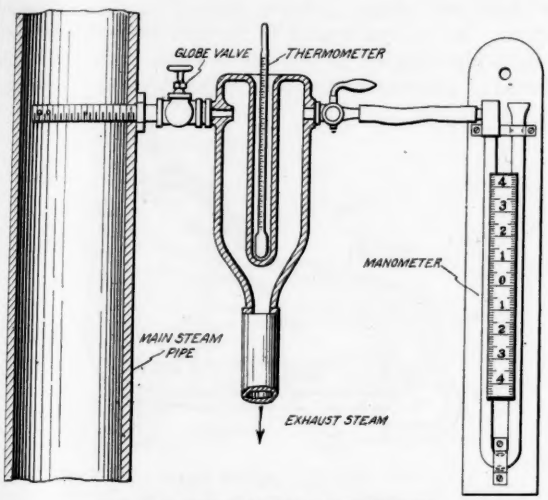


FIG. 7. THROTTLING CALORIMETER

speed results, while with large turbines the vibration often necessitates shutting down. Superheated steam lines, if long, may have water in them. Lack of pipe covering or of efficient covering is a certain cause of wet steam, as is also pipes which are too large. Branch pipes taken from sides or bottom of a main will have water in them; they should be taken from the top. Separators located near the machine using the steam will remove the moisture, and should be placed before the valves so that the steam will be dry when passing the valve seats.

Wet steam is a cause of rapid wear on valve seats and this is one way of finding out the presence of water. High water or priming in the boilers, long pipes, and improper drainage are other causes of wet steam.

Some of the effects of wet steam may be summarized: lowers efficiency of engines; causes leaks in pipes (rust holes, etc.); causes vibration of pipe; vibration of steam turbines and other machines; water hammer; corrodes valves, valve seats, and parts of engines, turbines, and other machinery; causes waste of oil; washes oil from valves and cylinder walls.

The remedies for wet steam have been suggested and include: covering for pipes; use of separators; drip pockets; proper drainage; slope of pipe; use of receivers; attention to boilers, to prevent priming.

The priming or amount of moisture carried by steam is determined by a calorimeter, of which there are several forms, throttling, separating and combined throttling and separating. When the moisture is not over 3 per cent the throttling calorimeter, Fig. 7, is used. A sampling nozzle made of ½-in. pipe is inserted in a vertical pipe. The nozzle should extend to within ½ in. of the opposite side of the pipe and have a closed end. Twenty or more ⅛-in. holes in the nozzle allow steam to enter it. Prof. Carpenter's design is shown in the figure. Steam containing a small amount of moisture will become superheated if the pressure is reduced by

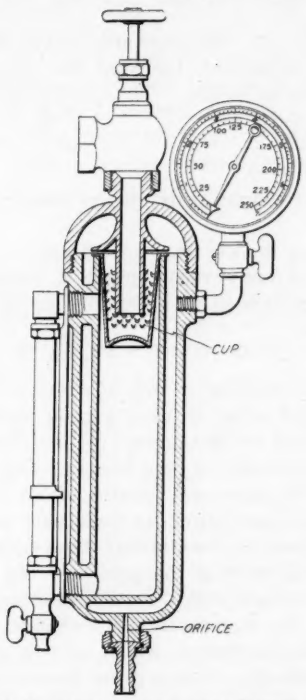


FIG. 8. SEPARATING CALORIMETER

throttling, without loss of heat. The readings to be taken are: pressure in steam line, pressure in calorimeter, temperature in the calorimeter. The quality of steam is figured by the formula

$$X=\frac{H_c-q+0.48\ (T_c-t)}{r}$$

in which

H_c = total heat corresponding to pressure in calorimeter

q = heat of liquid for steam line pressure

T_c = temperature in calorimeter by thermometer reading

t = temperature corresponding to pressure in calorimeter

r = heat of vaporization, latent heat corresponding to pressure in steam line.

All values except T_c are found from a steam table. Charts can be obtained giving the result without calculation.

The Carpenter separating calorimeter, Fig. 8, can be used when there is more than 4 per cent moisture. In this form the moisture is separated from the steam by passing through the corrugated openings in the cup. The steam passes out of the calorimeter through an orifice of definite size. The moisture collects in the inner chamber from which it may be drawn off and weighed for a known interval of time, or may be read from a scale

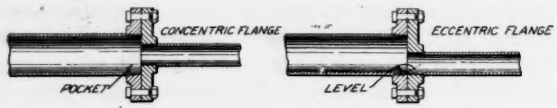


FIG. 9. CONCENTRIC AND ECCENTRIC CONNECTIONS

on the instrument. The gage attached to the calorimeter is graduated to indicate the total flow of steam through the instrument in 10 min. The quality of steam is

$$X = \frac{w + R}{W + w}$$

in which w = weight of dry steam passing out, obtained from gage.

W = water drawn from separator, or read on scale.

R = water due to radiation which according to Prof. Carpenter amounts to about 1/5 lb. per hour.

Arrangement of Drips

Improper draining or lack of provision for necessary draining is the cause of many piping troubles. A complete survey of the arrangements for draining will include a consideration of: the location, size, and length of drip pipes; the sizes and lengths of all piping and the manner of making changes in direction; the location and kinds of valves; the location of dead ends; the position of supports to see that the pipe does not sag; the location of all fittings; the direction and amount of pitch of the pipes; the manner in which changes in size of pipe are made; the manner in which branch pipes are taken off; the possibility of drip pipes becoming clogged up; the location of drip discharge valves; the location of

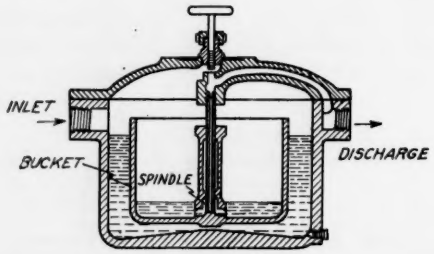


FIG. 10. BUCKET TRAP

drip pockets; the disposition of the water of condensation; the location and operation of steam traps; the location, sizes and condition of all drip piping apparatus, etc.

The damage caused by water hammer is well appreciated as is the danger due to a slug of water entering a steam cylinder. All low points in a piping system should be drained. Pockets where water can collect may be due to sag of a pipe; to position of a valve; kind of valve; direction of piping; changes in size, etc. Drip pipes should be taken from the bottom of the pipe, long pipe lines having drip pockets every 75 to 100 ft. In general, drip pockets should be the full size of the pipe as water is often carried across small openings. From the drip pocket, connection is made to a steam trap.

Drip pipes from main lines should be ¾ or 1 in. diameter; for long pipes of large size, 1¼ or 1½-in. drip pipe may be necessary. There is always the possibility of small pipe becoming closed up. The burr left after cutting will reduce the size of the pipe and often serves to hold dirt, scale, etc., until the pipe is closed. All drip pipes should be tested frequently to make sure that they are clear. It should be possible to know that such pipes are open and a sight discharge should be arranged for.

The fewer valves in drip lines the better. Such valves as are used should show plainly whether they are open or shut and should not be located in out-of-the-way places. All valves located in vertical pipes should have a drip pipe tapped in just above them. All dead ends should be drained, and should be provided at the bottom

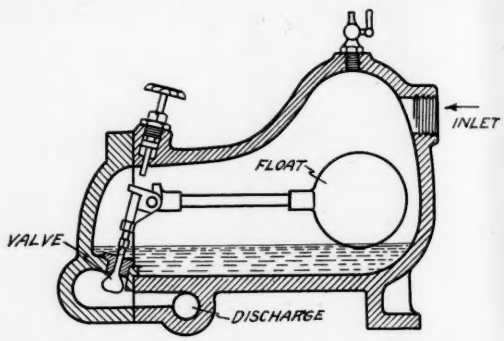


FIG. 11. FLOAT TRAP

of vertical pipes. Eccentric fittings can be used to avoid pockets when reducing the size of pipe, as shown in Fig. 9. For separators, traps, etc., the sizes of pipes are given by the manufacturers.

Water from steam main drips can be gathered in a receiver and automatically pumped back to the boilers. Supply pipes should slope from the steam header to separators, the slope for draining being about 1 in. in 10 ft. The supply pipe for the engine or turbine is then taken from the top of the separator. In this way slugs of water, scale, dirt, etc., are caught and the moisture is removed from the steam before entering the steam using machinery. It is important that the drips be sufficient in number, so located that water cannot collect and that all dead ends be drained.

Traps

The purpose of a steam trap is to dispose of the condensed steam from a piping system by discharging the water without allowing steam to escape. If a trap is not in good condition, steam may be escaping all the time which means a continuous invisible loss. Steam traps in order to give satisfactory service, require regular and constant attention. They should be inspected at least

once a day, and the piping should be arranged so that the discharge can be seen as well as heard. If necessary, an outlet can be taken from the discharge pipe to serve the double purpose of indicating the operation of the trap and showing whether steam leaks through the trap.

When the water from a trap discharges against atmospheric pressure, as into a hot well or sewer, it is a non-return trap. When the hot water is discharged back into the boiler, it is called a return trap.

The Walworth bucket trap, Fig. 10, collects the water of condensation until it overflows into the bucket. The bucket then sinks, uncovers the opening in the spindle and allows the water to be driven out the discharge pipe. The McDaniels float trap, Fig. 11, collects the water of condensation until it raises the spherical float which opens a valve and allows the water to be discharged.

The Cranetilt direct return trap, Fig. 12, collects the water of condensation which enters through the inlet check valve, 1, and passes through the divided trunnion tee into the tank, 2. When the tank fills, the weight of

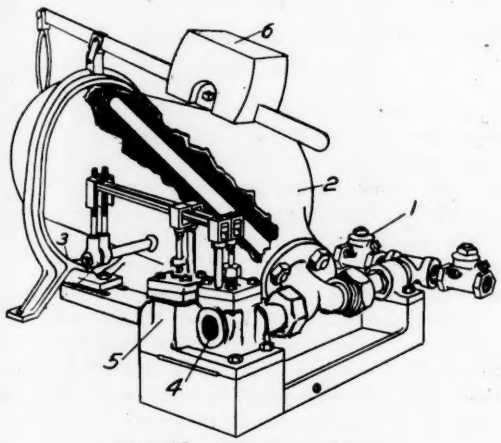


FIG. 12. CRANETILT TRAP

water causes it to drop to the bottom of yoke, 3, opening steam valve, 4, and closing vent valves. This allows pressure to enter through the inner pipe into the space above the water, and closes the inlet check valve. The pressure in the trap will then be the same as in the boiler, and as the trap is located above the boiler, the water flows to the boiler by gravity. The counterweight, 6, brings the tank back into the filling position after a sufficient amount of water has been emptied.

The size or capacity of a trap should be specified by the smallest number of pounds per hour which it will handle under the available steam pressure or the size and length of pipe which it will drain. Sometimes a trap is installed with a capacity just sufficient to take care of the ordinary water of condensation. Such traps fail, if a slug or larger quantity of water comes over, with consequent damaging results to engine or steam using apparatus. The sizes of pipes do not determine the capacities of steam traps.

Non-return traps should be located so that the water of condensation will flow to them by gravity in order to make sure of completely draining all pipes. Drip pockets and separators should be used to collect the water and drip pipes run from them to the steam traps. If possible, traps should always be located so that they will be easy to get at for repairs or adjustment. If the pressure is more than 5 or 6 lb. the trap can be located above the lowest point in the system to be drained if more convenient. Drips from pipes under different pressures should not be piped together.

Direct return traps must be placed at least 3 or 4 ft. above the water level in the boiler. The discharge pipe should pitch toward the boiler and should not have any other connections, and full boiler pressure must be supplied to the trap. Figure 13 shows the setting for a Cranetilt direct return trap.

In general, simplicity is desirable in steam traps and they should be constructed so that it is easy to get any working parts or to renew parts which become worn. One source of trouble is defective valves, due to scale, dirt, etc., which wear the valve seats. A strainer in the pipe to the trap will protect the valves or other parts and lead to more satisfactory operation. The working parts of a trap should be kept cleaned, and lubricated when possible. Sticking or binding and friction of joints and working parts are other causes of unsatisfactory operation.

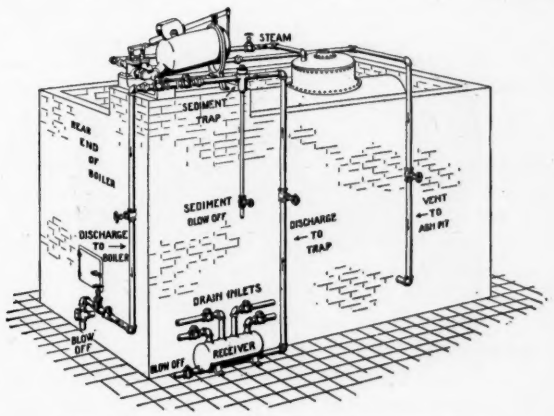


FIG. 13. SETTING FOR DIRECT RETURN TRAP

Buckets and floats should be tested for buoyancy to make sure that they do not leak. The collection of sediment in buckets is another source of trouble. The valves of traps which operate continuously become worn and should be renewed as frequently as necessary. Steam traps of whatever type must be kept in perfect condition all the time, if they are to serve their purpose well.

Overhauling of the piping is something which should be going on at all times. The purpose of piping is to get the steam to the place where it is used with the smallest loss, and with safety, and to do this continuously. For this reason repairs and changes should be made as soon as the necessity becomes known.

Neglect of repairs on piping, neglect of inspection of piping apparatus, continuance of improper drainage, continuance of too large a drop in pressure, leaky valves, uncovered pipes, and similar conditions are sources of constant loss and constant danger.

Overhauling and Testing Fans

For Leaks, Bad Arrangement and Performance. By M. S. Leonard

THIS ARTICLE is not to give a technical exposition of fan testing as conducted in a laboratory, but rather to describe as concisely as possible, tests and observations that can be made upon fans and their connections after installation, which will be of practical use in improving the efficiency of the useful results obtained.

The reader should realize at the outset that many fans are so situated, owing to complicated flues or ducts that very accurate measurements of the velocity and pressure cannot possibly be obtained. In such cases several determinations should be made at different locations, and by different methods. If the several results are then averaged after eliminating any that is very divergent, a fairly accurate measurement will probably be secured, and if this agrees with the known characteristics of the fan, it may be accepted as correct.

Inspection and Cleaning

Unless, for purposes of comparison with improved conditions later, it is desired to know what results are being obtained before any steps are taken to better them, all apparatus should be carefully inspected and put in good order before making any tests. If copies of the drawings of the fan and curves or tables of its performance are not already on file, it is well to obtain them from the makers. The interior of the fan, and all passages through which the air flows should first be thoroughly cleaned. In new work, refuse construction material or other things which do not belong there, are often found in the fan or ducts. In old installations foreign material often accumulates on the fan blades and in the casing and flues. In induced draft fans, soot and cinders, if not removed frequently, may accumulate in the bottom of the fan casing up to the periphery of the wheel and in the flues to an extent which reduces the area sufficiently to increase the friction losses appreciably, and to prevent the proper closing of dampers. Soot accumulating on the blades of fans having narrow curved blades, may fill up the depressions in the blades enough to affect the output and the efficiency seriously. Where forced draft fans are located where the air coming to them carries dust and ashes, especially if the inside of the fan is oily from overfilled bearings, a similar condition often exists. Sometimes underground ducts are partly filled with water. All of these things should be looked for and corrected, if they exist.

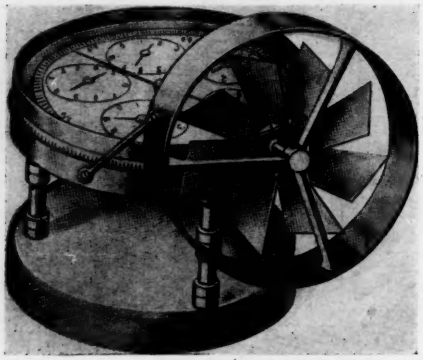


FIG. 1. PORTABLE ANEMOMETER

The mechanical and running condition of the fan should be inspected in detail. See that there are no leaks in the seams of the casing or between casing and flues and in so-called ¾ housing fans between casing and foundations. If bearings have not been periodically drained and cleaned, do so now, and see that they are properly alined and fitted, that they run free and cool and lubricate properly, and that the shaft journals are in good condition. In case of water-cooled bearings on induced draft fans, see that water circulates and keeps bearings properly cool.

Vibration

If all of these matters are in proper shape, and fan wheel is clean, the fan should run at any speed up to its safe maximum, without vibration or laboring. If it doesn't, the cause should be sought and removed. It may be due to an unbalanced condition of the fan wheel caused by injury to some of the blades by some solid object going through the fan, or by excessive heat, or by overspeeding at some time. Any large distortion can be seen. The vibration may be due to the shaft being sprung, or to the fan being out of line with the motor or engine which drives it, if it is direct connected. Remember that even a flexible coupling will cause vibration at high speed, if it is not lined up carefully. Uncouple the driver, and run it separately to see whether the cause is in it. Poor foundations are often the cause of vibration especially of units driven by reciprocating engines. If the cause of the vibration is not evident after investigation along these lines, the wheel should be rebalanced. Vibration developing in fans, and being neglected, is a common cause of breakdowns and repairs, and to some degree absorbs power and lowers the efficiency.

Bad Setting Up

Now compare the dimensions which affect the flow of air through the fan with the drawing of the fan. If fan has inlet cone, or cones, see that they come the correct distance from the wheel inlet. Also see that the inlet to the casing is not obstructed by anything the fan designers did not intend to have there. If fan has curved blades, see that curve has not been altered. See that the cutoff, that is the point where roundabout of casing approaches nearest the wheel, is in proper position. This is important in many types. Particularly in fans of the ¾ housing or ⅞ housing types, where a part of the scroll is contained in construction not built by the fan maker, make sure the form and dimensions are exactly as called for on drawing. Frequently, this work

is carelessly or ignorantly built, and is not at all as called for, causing a great reduction in capacity and efficiency. In connection with this, see that the connection from fan to flues is smooth and unobstructed. There should be no abrupt expansions and no contractions whatever.

In some cases, very likely, construction which is not correct will be found, where considerable difficulty would be experienced to change it. In such cases it is well to proceed with the tests without making the change, but find out from the fan manufacturers what the probable effect is, so as to be able to determine whether it would pay to change.

The piping systems should be gone over for leaks, and any that are found, repaired, and all dampers be made to close tightly, if they affect the volume handled by the fan.

When these things have been done, it is time to measure the work done by the fan and the power taken.

Fan Tests

In a fan test, the factor that it is most difficult to determine, accurately, is the volume, which is obtained

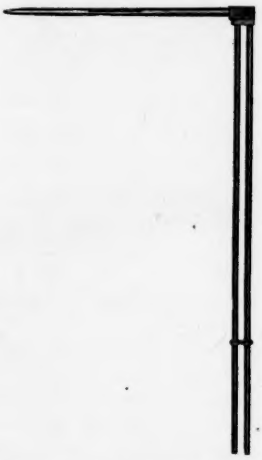


FIG. 2. COMMON FORM OF PITOT TUBE

from measurements of the velocity at some point. The instruments most frequently used for measuring the velocity of an air current are the anemometer, the pitot tube and the draft gage.

The anemometer, shown in Fig. 1, consists of a light and delicately constructed fan wheel, whose motion is transmitted to a system of practically frictionless gearing within an attached case. Upon a dial, the movement of the fan wheel is registered by hands revolving over graduated circles which read in total feet of flow. To use the instrument it is therefore necessary to take the time of starting and stopping. The anemometer is provided with a socket for attaching a long slim handle so that the operator does not obstruct the air current, and with a device for starting and stopping the recording mechanism by means of a cord.

Using the Anemometer

When all the air to be measured passes through one opening as a doorway, or the open end of a duct, which is accessible, and when the velocity is not too high for the instrument, the velocity may be measured by this means with a fair degree of accuracy. Ordinary anemometers are only good up to 20 or 30 ft. per sec., but some instruments are made that are claimed to be satisfactory at considerably higher velocities. The procedure is as follows:

Divide the area of the opening in which the measurements are to be taken into a number of equal areas, the more the better, by fine string or wires. Hold the anemometer in the center of each area successively and take a one-half minute reading, noting the reading of the meter dial at starting and at stopping for each area. The starting reading subtracted from the stopping reading, corrected for the error of the instrument, is the linear feet of flow during the time of observation, and this quantity divided by the time of observation, in minutes, is the average velocity through that area in feet per minute. The average of the velocities obtained for the several areas is the average velocity for the entire opening. This average velocity multiplied by the total area of opening in square feet, is the cubic feet per minute being moved. No two anemometers read alike and the instruments are also so delicate that they are easily disarranged by rough handling, especially if transported, so that it is customary to have them calibrated frequently; hence, the reference to instrument error above.

The practice of taking one continuous reading, moving the anemometer back and forth over the area of the entire opening, as is sometimes done, leads to an error on the side of too great volume when the velocity is not the same over the whole area, because of the momentum of the fan which does not slow down instantly when moved from a higher velocity to a lower one. The anemometer method of measuring volume is not recommended when the pitot tube method can be employed.

Pitot Tube

The pitot tube, one form of which is shown in Fig. 2, does not measure velocity directly, but enables the pressure head corresponding to the velocity to be indicated upon a manometer or draft gage two forms of which are shown in Figs. 3 and 4, from which the velocity can be calculated.

As air or gas flows from place to place through flues or ducts, its pressure is constantly changing. The pressure at any given cross section of the conduit is made up of two components, the static pressure and the velocity pressure. The static pressure measured with reference to the atmospheric pressure, if above atmospheric, is the pressure being used to overcome the resistances to flow between the point of measurement and the atmosphere. If below atmospheric, it is the pressure to overcome resistances to flow between the atmosphere and the point of measurement plus the pressure to produce the existing velocity. This difference in the significance of the static pressure when above and below atmospheric, is important and should not be lost sight of.

Velocity pressure in both cases is the pressure due to the impact of the moving gas by virtue of its weight and velocity. The static pressure is exerted in all directions, the velocity pressure only in the direction of the

flow. Consequently a pressure indicating device connected to an orifice facing directly upstream will indicate the algebraic sum of the two pressures, while if connected to an orifice facing exactly at right angles to the stream it will indicate the static pressure only. As a matter of fact, the static orifice must be of a certain form to transmit the pressure correctly. Most orifices produce an error when there is a velocity past them due to the induction of the moving stream which makes the indication lower than the true static pressure.

The pitot tube is an instrument combining in a convenient form two tubes in such a position that when the orifice of one faces the stream those of the other are at right angles to it and the orifices of the second tube are of a size and shape to prevent induction effects. Two kinds are in general use. Both are substantially the same except for the form of the static openings. The Taylor pitot tube used in testing most of the fans purchased by

FIG. 3. CONVENIENT FORM OF U TUBE DRAFT GAGE

the United States is shown in Fig. 2. The single member projecting to the left, which is the end that receives the pressures, consists of two concentric tubes, the inner one open at the end and in communication with the right hand connecting tube, and the outer one, closed at the end by a tapering piece which does not disturb the flow, which has two long narrow slots communicating with the annular space between the two tubes, which communicates with the left hand connecting tube.

The pinhole pitot tube, so called, is the same except that it has four small "pinholes," two on each side in place of the slots. Most commercial fans are rated on the basis of tests using this form of tube.

Draft Gages

Water gages for indicating the pressures are of many forms but all are essentially U tubes either with both legs of the same diameter or with one leg so much greater in diameter than the other that the entire volume of the lesser bore does not appreciably alter the level in the greater. If the U tube is partly filled with liquid, the difference between the pressures impressed upon the surfaces of the liquid in the two legs will be indicated by the difference in the level of the liquid in the two legs.

For ordinary testing, only two simple forms will be found necessary, the plain vertical U tube with scale attached shown in Fig. 3 for pressures of an inch of water column or more and the inclined differential gage shown in Fig. 4 for smaller pressures. This type of gage usually uses an oil of a certain specific gravity, instead of water, but the scale reads in inches of water.

In addition to the pitot tube and draft gage sufficient rubber tubing is required of good quality and perfectly airtight to reach from the point of measurement to the draft gage and a Y connection for the same. Also a good thermometer to measure the temperature of the gas. Some pitot tubes have solder used in their construction. Such instruments are obviously unsuited for use in temperatures which will melt the solder.

Connection Arrangement

To determine the volume flowing in a duct by the pitot tube method, the procedure is as follows. Select

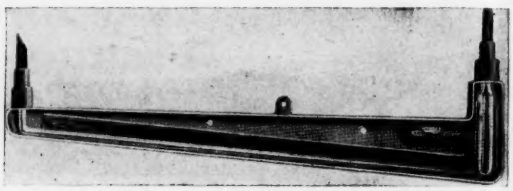


FIG. 4. ELLISON DIFFERENTIAL DRAFT GAGE

a place where the flow is likely to be as undisturbed as possible. There should be a length of at least 15 diameters of straight flue upstream from the point of measurement. Attach two draft gages to a firm support in a convenient location. If the differential gage is used, and it usually should be for velocity pressures, be sure it is set level in both directions. The impact side of the differential gage is at the upper end of the inclined leg. Connect first one side and then the other of the pitot tube to the impact side of the draft gage with the rubber tube and test for tightness by blowing up considerable pressure and trapping it by closing the openings in the pitot tube. If the pressure does not drop it shows that pitot tube and tubing do not leak. If the pressure does drop there is a leak which must be eliminated to insure correct readings.

Connect the impact side of the pitot tube to the impact side of the draft gage that is to be used to measure the velocity pressure and the static side of the pitot tube to the other side of the velocity draft gage, with a Y connection in the tubing. Connect the branch of the Y to one leg of the other draft gage. Figure 5 shows the arrangement. It is evident that the gage to which both sides of the pitot tube are connected indicates the difference between the pressures impressed on the two sides of the pitot tube, that is the difference between the impact pressure which is the sum of the static and velocity

pressures, and the static pressure alone, or it indicates the velocity pressure direct. The other gage indicates the static pressure.

Readings and Calculations

Provide openings through which to insert the pitot tube into the duct, so located that readings can be taken in the centers of numerous equal areas over the entire area of the duct. In the case of round pipes the readings are usually taken across two diameters at right angles to each other the locations being at the centers of zones of equal area. The positions can be determined before readings are taken and marked on the shank of the pitot tube. Before cutting more than the first hole in the duct it is well to explore the whole area with the pitot tube through the first hole, observing the indication of the static gage. In order that the velocity pressure readings be correct, the static pressure must be the same all over the duct. If it is not, obviously there must be flow from points of higher to points of lower pressure, transversely of the duct, which means that the flow is not in straight lines parallel to the sides of the duct and so the velocity cannot be used to obtain an accurate volume. If the static pressures are not very nearly the same it is best to try to find another location where they are.

Record the zero reading of the impact gage for each observation. This is better than trying to set it at actual zero and serves as a check on any movement of the gage or loss of liquid. Traverse the pipe and record the velocity head at each predetermined point. The pitot tube should be held pointing directly upstream. This is when the gage reading is a maximum if the flow is parallel to the sides of the duct. Compute the velocity for each reading from the formula

Velocity, ft. per minute = 1097 times the square root of (Gage reading in in. of water divided by wt. of one cu. ft. of gas.)

The weight of 1 cu. ft. of gas is obtained by multiplying the weight of 1 cu. ft. at 32 deg. F. and pressure 29.92 in. by the proportion obtained from the expression

$$\frac{492}{460+t} \times \frac{P+29.92}{29.92}$$

in which t is the temperature of the gas as observed and P the observed barometer pressure in inches. A thermometer should be inserted in the duct near where readings are taken and its indication recorded frequently for this purpose.

The average of all the velocities thus obtained is the average velocity in the duct and this multiplied by the area of the duct in square feet is the volume passing in cubic feet per minute.

In boiler practice when it seems to be impossible to get reliable volume readings by either the anemometer or pitot tube method, resort can be had to calculation from the amount of fuel burned, the analysis of the fuel and the analysis of the gas at the fan.

Work of Fan

To determine the work done by the fan, it is necessary to know in addition to the volume handled, the total pressure imparted to that volume by the fan. This is obtained by means of the pitot tube and water gage, in a manner analogous to the determination of the velocity pressure, using only the impact side of the pitot tube connected to one side of the draft gage. The average impact pressure in the duct leading to the fan is measured and the average impact pressure in the duct leading from the fan is also measured. These readings should be taken as near the fan as it is possible to get steady gage readings. The difference between these two average pressures is the pressure produced by the fan and the work done by the fan is obtained by the formula (for small pressure differences)

$$\text{Air horsepower} = \frac{Q\,h}{6345}$$

where Q is the volume in cubic feet per minute and h is the gain in impact pressure in inches of water column.

When the impact pressure readings are not taken right at the fan, the friction loss between the fan and points of reading should be calculated and added to the pressure gotten by subtracting the impact readings, because the pressure loss between the two points of reading

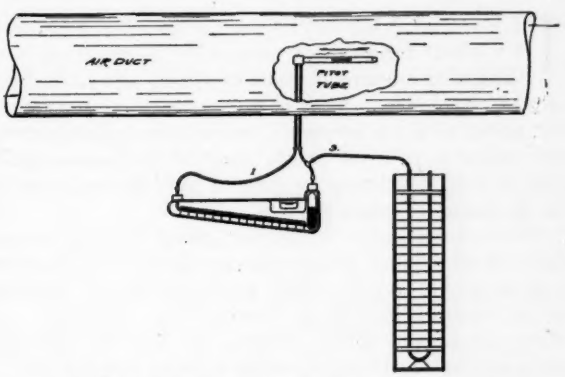


FIG. 5. USE OF PITOT TUBE AND DRAFT GAGES

is not included in the readings, but is included in the pressure produced by the fan.

The formula for friction loss is

$$\text{Friction loss in in. of water} = \frac{kSV^2}{A}$$

where S is the rubbing surface in square feet; V is the velocity in feet per minute; A is the area in square feet and k is a constant depending on the nature and condition of the surface. Its value for clean metal flues may be taken as 0.000,000,000,4 and for brick flues as 0.000,000,000,8.

The air horsepower divided by the horsepower input to the fan gives the fan efficiency.

The methods of determining by test the horsepower of all of the various types of machines used for fan driving is not within the scope of this article, but in most cases they are probably better known than the methods of measuring the work done by the fan and their capacities can usually be obtained from the makers. The only instruments necessary, indicators, pressure gages and speed counters for steam machines, and ammeters, voltmeters, and wattmeters for electrical machines, are nearly always at hand, and their use is understood.

It is well to compare the result of the fan test with the fan manufacturers' tables or curves of capacity. If the results fall on the curves or tables, it is an added assurances of their correctness. If they do not, an explanation should be sought, because the data of the reputable manufacturers are obtained under more favorable conditions than prevail in almost any installation, and are entirely reliable.

In conclusion, the writer would like to point out that usually greater economies can be obtained by improvements in the system connected to a fan than by any change in the fan itself, unless it is much unsuited to its work.

The elimination of leaks either into or out of the system or through dampers is often a great saver of power. These can be found by volume measurements or gas analyses at successive points.

The elimination of unnecessary pressure losses due to constricted areas at dampers and elsewhere, and to abrupt changes of direction or velocity, can often be accomplished at slight expense and show a large power saving. Such bad points can be found by observing the drop in pressure from point to point along the duct and where a sudden large drop occurs, seeking the cause.

Where several fans are used on the same system, economy often results from a study of the best combination to operate for a given load. A study of the air requirements for various conditions and of the fan performance curves will result in running such apparatus as will operate at or near its best efficiency at all times. The fact is frequently lost sight of that a fan may be so lightly loaded as to be very inefficient and sometimes a saving of power can result from shutting down a fan and letting the others take its load.

Keeping the Steam Engine Tuned Up

LOOKING AFTER THE VARIOUS DETAILS. TESTING FOR LEAKAGES. INDICATOR DIAGRAMS. CORRECT RECEIVER PRESSURE. BY GEORGE S. BLANKENHORN

THE STEAM ENGINE is so old and has been written about for so many years that it would seem as though to attempt to write anything new on it were next to impossible. What is necessary to the upkeep of this prime mover is generally understood, but the writer will endeavor to cover briefly some of the points which may be overlooked even by some of the best engineers in the operation of steam engines.

We will first cover briefly the upkeep of the principal parts of the engine, going into the inspection, operation and remedy of defects which are often found. Second, we will take up indicating, valve setting, adjusting, regulation, etc., and joints which make for the bettering of operating conditions, from a thermal standpoint.

BEARINGS

THIS part of any prime mover needs more attention than any other and it is especially true of the steam engine. No matter how sure the operating engineer is of his positive, or other type of oiling system, he should at short intervals inspect and feel each of the important bearings on his machine. Now and then he will notice that a crankpin, wristpin, or main bearing has too much play. As soon as this is noticed, no time should be lost in keying up, or in the event of the main bearings, the quarter boxes should be adjusted by means of the wedges, or set screws, or by whatever means are provided.

In the case of the bearing with a spherical seat, there is always a double wedge adjustment for both quarter boxes, and care and judgment must be used in taking up on each in such a way that the center of the shaft is not thrown out of the center line of the lower shell. If this course is not adhered to, the bearing will heat badly. Always use a feeler as much as possible when adjusting bearings. Likewise a feeler should always be used after adjusting the crosshead shoes. The clearance between the upper shoe and guide should be about 0.007 in. It is sometimes advisable to use an inside caliper, or preferably a pin gage, to see that the crosshead shoes are so adjusted that the piston rod is parallel with the lower guide. This should be done with the crosshead on the crank end dead center. A bearing which has just been taken up should be watched very closely after starting for the first time, to see that it does not overheat.

ALINEMENT

IF AN engine is out of line, it is usually shown by some of the bearings heating; as, if one main shaft bearing is ahead or behind the other, or higher or lower, the crankpin bearings would first show distress, then possibly the main bearings. A good level placed on the main shaft may be used to check up the bearings for height. If the indications are that the two main bearings are not in line, they should be checked by removing the main bearing cap and one quarter box, and by the use of a pin gage between the center of shaft and frame jaw, used first on one end of the bearing and then on the other.

The only way to determine whether the cylinders are in line with the frame is to remove the piston, piston rod, connecting rod and crosshead, and run a line through the cylinder and ways. In the case of a compound engine where the foundation has settled, or is believed to be out of line, it is best to pull the rods, shaft, etc., out and reline, using lines, pin gages and levels.

OIL SYSTEM

THERE are so many types of oiling systems that we will only touch upon some of the important points to look out for, which might apply to any of them. The oil pump valves should be inspected occasionally, as often small pieces of waste and other foreign matter lodge between them and their seats. The piping should be watched for stoppages; any portion of this is readily taken down on modern installations without interfering with any other part. If an oil filter is used, this must be cleaned out and new filter bags put in, water drawn

off, etc. The different bearing sight feeds should be carefully scrutinized several times a day, as often foreign matter will lodge between the needle valve and its seat and cut down the oil flow.

Valves

Valves of a Corliss do not need much attention, but they must be kept tight and may be tested in the following way:

Taking first the head end steam and exhaust valves, place the engine on the crank and dead center, close both steam valves (simply unhook and the dashpots will close them) disconnect the connection between the head end exhaust valve and wristplate, and block this valve so that it will remain in the closed position. Now open the head end indicator cock and slowly open the throttle. If the steam valve leaks, steam will blow through the open cock. To test the head end exhaust valve, open the head end steam valve, close the indicator cock and open throttle slowly to build up pressure, then from some point in the exhaust line inspect for steam passing.

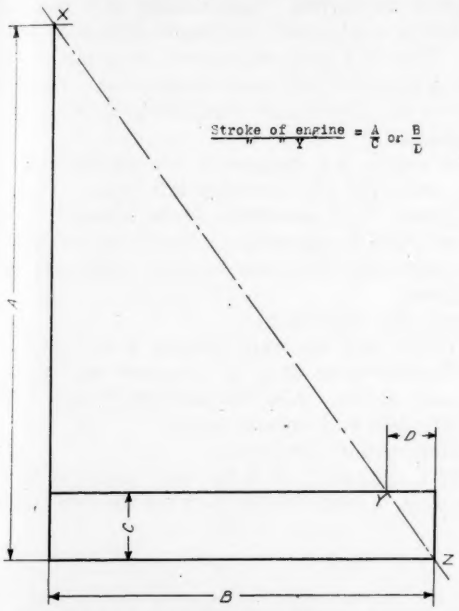


FIG. 1. SKETCH OF PANTOGRAPH REDUCING MOTION. X, Y AND Z MUST ALWAYS BE IN A STRAIGHT LINE. FOR A REDUCED STROKE OF ENGINE, SHORTEN A AND B, UNTIL WITH THE SHORTER MOTION OF X, Y TRAVELS THE SAME DISTANCE AS BEFORE, 4 IN., KEEPING X, Y AND Z IN A STRAIGHT LINE

To test the crank end valves, place the engine on the head end dead center and repeat the above. In the case of a compound engine, a small leakage in the high pressure cylinder exhaust valves is of no consequence, as this steam passes to the receiver and is used in the low pressure cylinder.

Pistons

These are of either one of two types, the box, or the built-up, the small size engine using the former. The rings of the box piston can be inspected only by removing the piston from the cylinder, but the built-up type may be thoroughly inspected by removing the follower, and then the junk ring. If it is deemed advisable to test either type of piston for leakage, this may be done as follows:

Remove the cylinder head, place the engine on the head end dead center, close the head end steam valve and the crank end exhaust valve and open the crank end steam valve; then slowly open the throttle and build pressure up behind the piston. Any amount of leakage is then readily seen.

Valve Motion

These bearings must be well lubricated and kept free from play. The packing gland of the steam valve stem must not be kept so tight that the dashpots are unable to close the valves readily. The drop rods to the dashpots should be of such a length that when the pot is down and the hooks are at the low end of their travel, there is 1/32 in. clearance between the engaging hook plate and the steam arm plate.

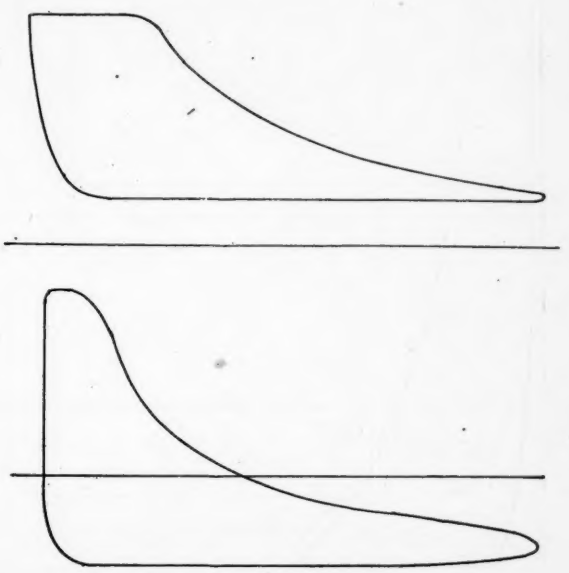


FIG. 2. CARDS FROM COMPOUND ENGINE, RECEIVER PRESSURE 28.5 LB.

From time to time it will be found necessary either to renew or to change the contact side of the hardened steel plates on the steam hook and steam lever arm. This is made necessary by the sharp corner finally becoming rounded due to continuous operation under rather severe conditions.

Governors

This very important piece of mechanism must be gone over occasionally to see that all parts are free, and now and then the pins must be renewed or there will be considerable lost motion, which is a great handicap to the proper functioning of the governor. If the governor is one of the late design, high speed, spring tension or compression types, with its own circulating oiling sys-

tem, it should run for long periods without attention, except for the changing of oil.

Indicating

This should be done at least every six months, and oftener if any peculiarities develop in the engine which cannot be detected by other means. If the engine is a compound, it is preferable to use an indicator on the high-pressure and low-pressure cylinders simultaneously; this is especially necessary when the engine is one carrying a fluctuating load. If only one indicator is available, this is far better than none, as even with a varying load it may be used first on the high-pressure, then on the low-pressure cylinder with a view to checking valve setting and other defects, but it is useless to determine the indicated horsepower when this is the case.

Most modern engines are sold with suitable indicator piping and a three-way cock for each cylinder. This three-way cock is made to take the standard indicator cock without additional fittings. Before attaching the indicator the piping should be thoroughly blown out. This may be done with the engine running, simply by opening the angle valves, then the three-way cock, first one way and then the other. Care should be used to see that the indicator piston is screwed on tight and it should be well lubricated with cylinder oil. After replacing the piston it is well to move it up and down several times to see that it does not bind. The next thing is to select the proper spring and place this in the indicator. In doing this, remember that the diagram must not be over 2 in. in height. For instance, if the pressure at the throttle is 100 lb., a spring marked 50 lb. would be suitable; for 150 lb., a 75-lb. spring should be used, and so on.

Two types of reducing motion are in general use, the reducing wheel and the pantograph. The writer favors the latter and believes that if engineers will use only the pantograph, they will find it more convenient, easier to handle, and more accurate. The writer has found that most operating engineers do not indicate often, if at all, due to the bother of rigging up the reducing motion; he will, therefore, show a sketch (Fig. 1) which gives the correct proportioning of a pantograph for producing a 4 in. card with various lengths of engine strokes. This is a very satisfactory reducing motion to operate, is accurate, and easily constructed. This should be made up of ¼ by 1¼ in. steel, using ⅜ in. cap screws at the joints.

If the engine is a compound, two pantographs must be built, one right and the other left hand. These may be supported from extension studs screwed into the frame, or if this is impossible, a simple support from the concrete or wooden floor may be used, which will answer the purpose.

Reasons for indicating:

1. To see that the valve setting is correct.
2. To determine drop in pressure through ports, throttle and valves. Also the amount of back pressure in the cylinders and exhaust lines.
3. Horsepower developed.
4. If a compound, it is the only means of knowing how the work is proportioned between the two cylinders.

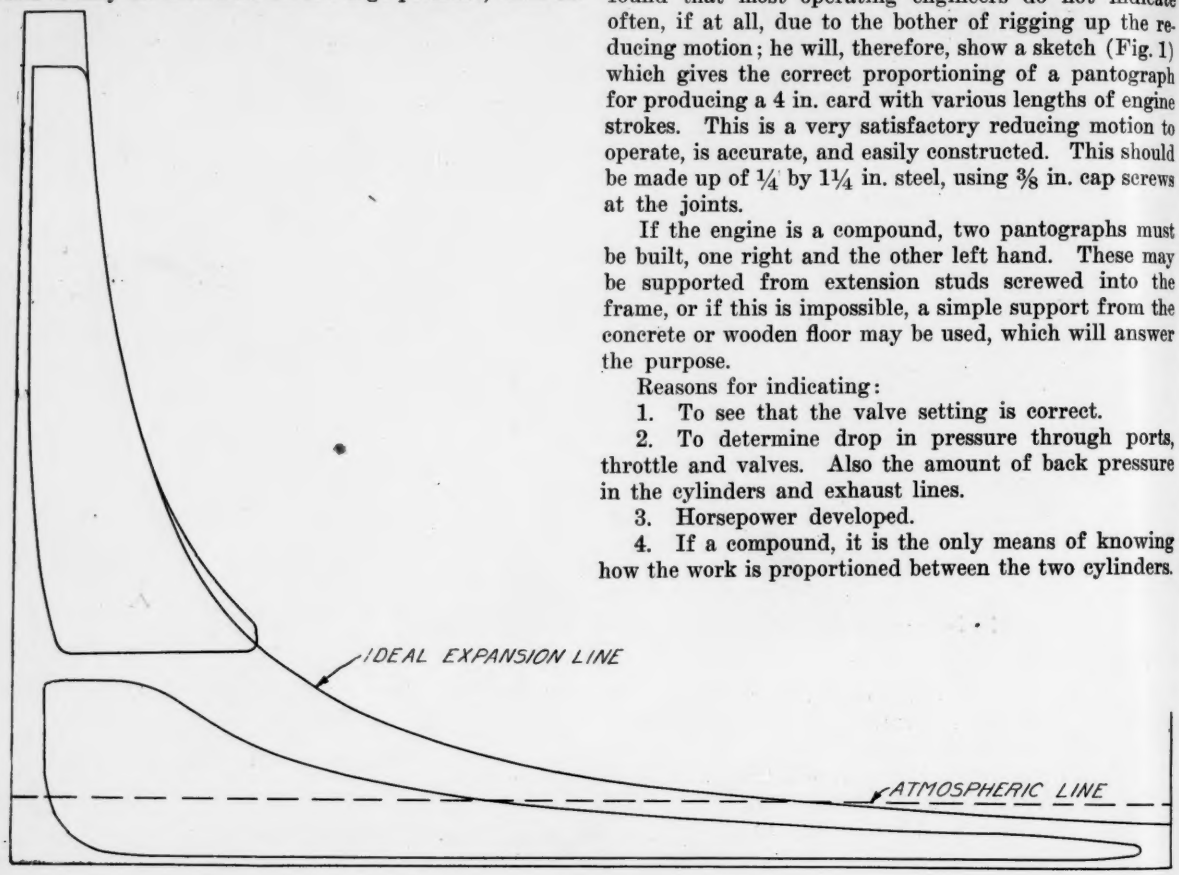


FIG. 3. CARDS OF FIG. 2 COMBINED

The writer will take the four reasons for indicating, as outlined above, and will discuss the first three briefly, but will go into the fourth in detail, due to its importance and the fact that it is usually overlooked.

1. The crank and head-end cards should have the same area, thus showing an equal amount of work in each end of the cylinder. The steam line should be parallel to the atmospheric line, or nearly so, and if the engine is high-speed, this line should fall off only slightly. Release should come slightly before the end of the stroke. Compression should start so that the final compression pressure is aproximately one-half the initial pressure. The admission line should be perpendicular to the atmospheric line. If it leans slightly toward the toe of

the diagram, it shows insufficient lead, and if the reverse is true, too much.

2. If we place a calibrated gage on the engine at the throttle and read it at the time of taking a card, we are able to determine the loss in pressure, due to wire-drawing through throttle valve and ports. It is necessary only to scale off the card pressure from the atmospheric line to the steam line. The difference between this pressure and that shown on the gage represents this loss, which in good practice should be from 3 to 5 lb. If more than that, it may be due to too small a steam separator, or throttle, or that the throttle is not wide open, or that the ports are not of sufficient size.

The back pressure in the cylinder, as shown by the card, should be within a pound of the exhaust pressure in the line, if non-condensing, and within half a pound, if condensing.

3. To determine the indicated horsepower being developed by an engine where the load is constant, three or four sets of cards should be taken. If the load is fluctuating, three or four times that number are necessary to get accurate results. If a compound is being indicated, high-pressure and low-pressure cards must be taken simultaneously.

In working up the cards, the area of each must be carefully determined either by planimeter, or by laying off ten ordinates equivalent and perpendicular to the atmospheric line. In the former case, after determining the area, divide it by the length of the card, which gives us the average height; this multiplied by the scale of the indicator spring is the mean effective pressure. If the ordinate method is used, add the 10 lengths together and divide by 10; this result multiplied by the scale of the indicator spring is the mean effective pressure. If the ordinate method is used, add the 10 lengths together and divide by 10; this result multiplied by the scale of spring is the mean effective pressure.

After the mean effective pressure has been found for both head and crank ends take the average and substitute in the formula given below to determine the indicated horsepower.

FIG. 4. CARDS WITH RECEIVER PRESSURE 17 LB.

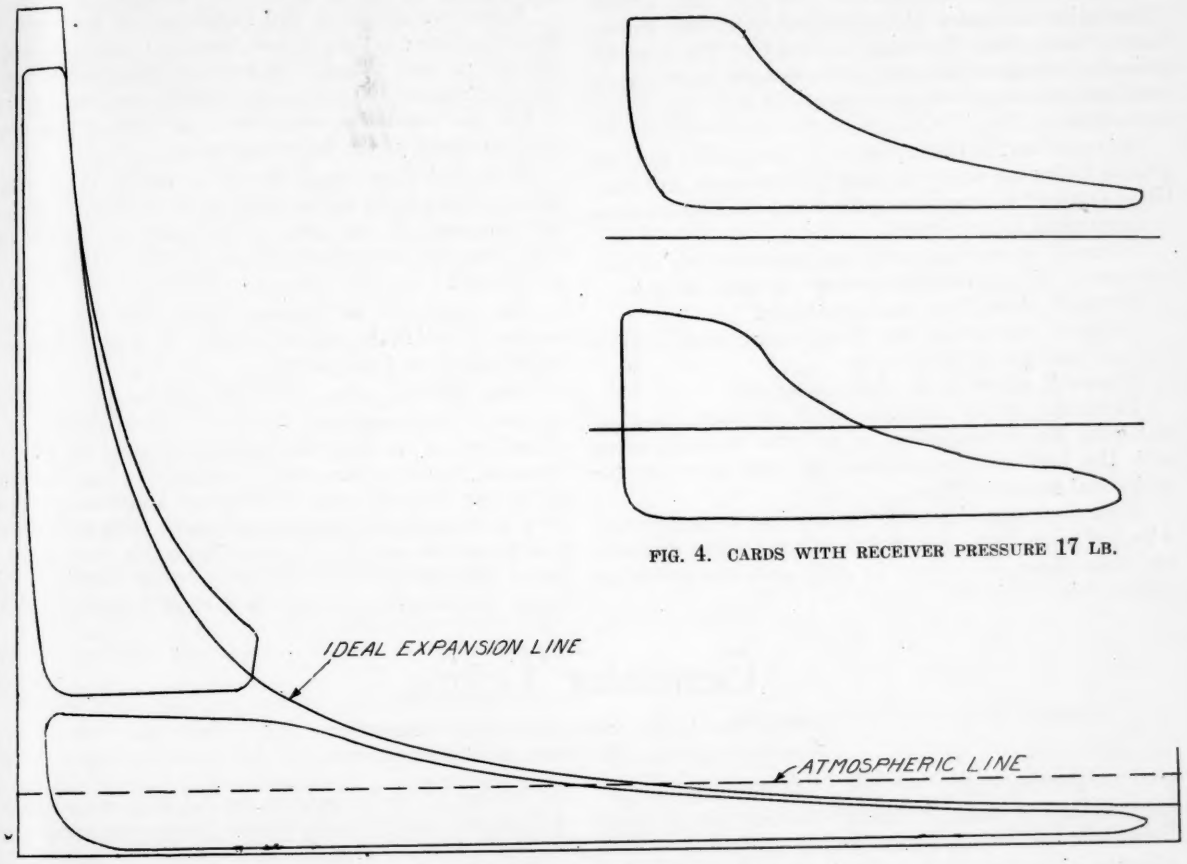


FIG. 5. CARDS OF FIG. 4 COMBINED

$$\text{I.hp.} = \frac{\text{Plan} \times 2}{33{,}000}$$

Where P = M. E. P. (average mean effective pressure).

Where l = length of stroke in feet.

Where a = area of piston in inches.

Where n = r. p. m.

4. Few engineers operating compound engines realize that there is one correct receiver pressure for an engine with their steam conditions (regardless of the amount of load). In a compound, if we want to change the load distribution between the high and low-pressure

cylinders, it is done by changing the low-pressure cylinder cut-off which changes the receiver pressure. Thus if we want to raise the receiver pressure, we shorten the cut-off of the low-pressure cylinder. To lower it, the cut-off of the low-pressure cylinder is increased.

Every engineer has heard something of the ideal indicator card, the card so constructed that the expansion curve is the isothermal for gas (PV = Constant), which means that during expansion the temperature of gas would remain constant. The more nearly the actual indicator card can be made to approach the ideal card, the higher the economy. The engineer who is operating a compound to get the best results should take a set of cards with various receiver pressures, and either combine them himself or have someone do it for him. (See article "Combining Indicator Cards" by the Author, Power Plant Engineering, February 15, 1918). The receiver pressure corresponding with the set of cards which, when combined, most nearly approaches the ideal, is the correct one to use.

So important is this question of the correct receiver pressure, that the writer is going to show cards and combined cards of a compound with which he had occasion to experiment quite recently.

Figure 2, shows high and low-pressure cards of this compound, with a receiver pressure of 28.5 lb. gage.

Figure 3, shows these cards combined.

Figure 4, shows high and low-pressure cards, with a receiver pressure of 17 lb. gage.

Figure 5, shows these cards combined.

From the actual indicator cards as taken, they all look good, but on combining, we find that the cards taken with the high receiver pressure do not approach the isothermal expansion line.

Tests were run with these two receiver pressures, and with the same high-pressure cut-off, and approximately the same steam conditions in each case, the following results were obtained:

Receiver pressure, lb. gage	28.5	17
Steam pressure at throttle, lb. gage	169	173.4
Vacuum at condenser, in mercury	27.5	27.9
Total i.hp. developed	148	162
Pounds dry steam per i.hp.	14.57	12.7
Pounds dry steam per hour	2179	2050

Summing up the above results, this engine when operated with 17 lb. receiver pressure used approximately the same total steam per hour and developed 9.5 per cent more power. This actual case demonstrates clearly the importance of operating an engine with the correct receiver pressure.

Speed Regulation

When we speak of the regulation of any prime mover, we refer to the per cent increase in speed between full and no load. Thus if we have an engine which with full load runs at 100 r.p.m., and with no load has a speed of 103, the regulation would be 3 per cent. This may be determined in the following way:

With full load, count the speed with a speed indicator. This we will say is found to be 100 r.p.m. Then, with no load, do the same. This speed we find to be 103; then the per cent regulation is $(103 - 100) \div 100 \times 100 = 3$.

The regulation of engines varies with the class of service for which the engine is built. It usually is found to be from 2 to 4 per cent.

From now on, more and more will be demanded of the power plant engineer, due to the ever increasing cost of fuel and at the same time greater demand for power. He must, therefore, keep every part of his plant working at the very highest point of efficiency, and it was with a view to helping him get better results with his engine that the writer was asked to contribute this short article. Space does not permit of going into great detail, yet he hopes that some benefit may be derived from his efforts.

Generator Testing

Inspection, Input and Output Measurements and Regulation. By Ottomar H. Henschel

IT IS IMPORTANT TO SEE that the bearings and journals of dynamo electric machines are in good mechanical condition, in proper alinement and so adjusted as to provide a uniform air gap at all points along the periphery of the revolving member. A wooden wedge of the form shown in Fig. 1 and graduated in the manner indicated will prove extremely useful in the checking up of air gap widths.

When starting a new machine, or one having been out of service for some time, remove the top bearing caps and raise the rotor or armature sufficiently high to give access to all parts of the shaft and lower half of the bearing. After cleaning well with a piece of cloth saturated with kerosene, carefully wipe these parts so as to remove all traces of dirt, grime or other substances which may have a tendency to cause heating.

If the oil well contains old or dirty oil, remove this and fill with clean kerosene to such height as will bring the surface of the kerosene in contact with the oil rings. After a half hour or more, remove the kerosene and wipe the well out thoroughly.

Use only a high grade of oil, and when filling the well pass the oil through a strainer, bringing it up to such a height as will insure a copious supply to the shaft, with the rings working properly.

Heating may be generally attributed to any of the following causes: (a) sticking of oil rings; (b) excessive belt tension; (c) rough bearing surface; (d) improper alinement of bearings or fitting of journal boxes; (e) lack of end play.

In the event of heating, feed the bearing an abundant supply of heavy oil, loosen the top bearing caps and, if belt driven, slacken the belt. Should the heating under this treatment not subside, shut down, care being taken, however, to prevent "freezing" by keeping the machine running slowly until the shaft is cool.

Frequent inspection of the bearings is advisable, especially in the case of high-speed machines.

Brushes and Commutator

To prevent "grooving" of the commutator, the brushes of one set should always be placed in a position staggered to those of the set following, and to insure proper commutation, the various sets of brushes must be spaced equidistant about the periphery of the commutator. To check this, measure the distance between the heels or toes of the various sets of brushes by inserting below the brushes a strip of paper about 1 in. wide, and equal in length to the circumference of the commutator and having placed upon it equally spaced pencil marks equal in number to the number of sets of brushes employed. By alining the toe of one brush with any one of the marks on the strip of paper, the toe of each of the other brushes ought to coincide with the remaining marks. If not, adjust the faulty sets until this condition is realized.

Attention should be given the pressure of the brushes upon the commutator. Brush tension may range from a minimum of 2 lb. per square inch to a maximum of 5 lb. per square inch, depending upon the characteristics of the brush, and the speed and mechanical condition of the commutator. Generally, however, tensions of 2.5 to 3.5

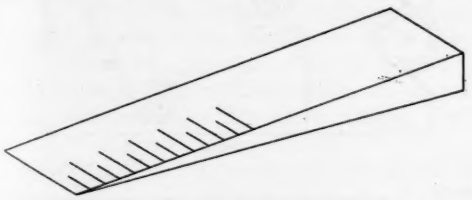


FIG. 1. GAGE FOR CHECKING AIR GAP

lb. per square inch are used on commutators, and 3 to 5 lb. per square inch on slip rings.

Brush tension may be conveniently determined by means of a small spring scale in the manner indicated in Fig. 2.

To provide proper contact surface between the brushes and the commutator, insert a strip of sandpaper, with the rough side toward the brush face, between the brush and the commutator, and with the brush in position, draw the sandpaper in the direction of rotation, repeating this as often as necessary to secure a smooth face; do not allow the sandpaper to come in contact with the brush during the return strokes.

Commutators, at all times, must be maintained as smooth as possible, and if unduly worn so as to cause excessive sparking, turning down should be resorted to. After this has been done and the entire surface given a smooth finish, the commutator should, within a short time, provided it is receiving the proper care, acquire a highly polished surface.

High mica is a common source of sparking, but may be avoided by undercutting.

Connections and Windings

Each time the machine is shut down, all accessible binding posts and connections should be carefully examined and tried out and if found loose, should be tightened or resoldered, as the case may be. The windings ought likewise be looked over and, if dirty, should be blown out and wiped off.

Measurement of Input

Where the driving unit is a steam or internal combustion engine of the reciprocating type, the amount of energy delivered to the generator may readily be determined from indicator cards. By taking a card first with the set running unloaded, and then another under load, calculating the number of horsepower developed in each case and taking the difference, we secure the total number of effective horsepower. To reduce this to kilowatts, multiply by 746 and divide the product thus obtained by 1000.

The indicated horsepower of a reciprocating steam engine is obtained by dividing the product of the number of pounds of mean effective pressure, the length of stroke in feet, the area of piston in square inches and the number of strokes per minute, by 33,000, the various factors

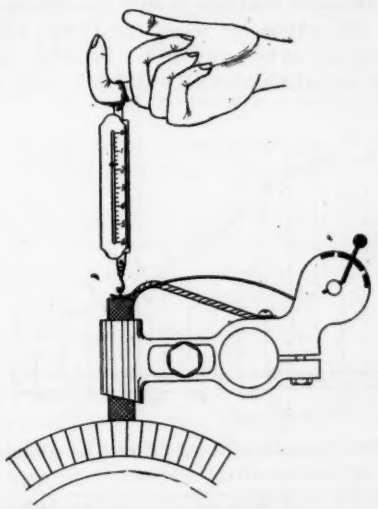


FIG. 2. METHOD OF CHECKING BRUSH PRESSURE

involved, except the mean effective pressure, being secured by measurement and count. By employment of the method of ordinates or the use of a planimeter, the mean effective pressure may be readily obtained from the indicator cards.

To use the method of ordinates, erect a series of equally spaced ordinates perpendicular to the atmospheric lines, as indicated by aa′, bb′, cc′, etc., in Fig. 3. Midway between these erect a second set of ordinates as 1′1, 2′2, 3′3, etc. The average of the sums of the lengths of these numerically designated ordinates measured in inches and multiplied by the scale of spring employed will give, as a product, the mean effective pressure in pounds per square inch.

If the machine is of the belted type, the number of horsepower transmitted by the belt is approximately obtained by dividing the product of the velocity of the belt, in feet per minute, and the width of the belt, in inches, by 550. As in the foregoing instance, this may be re-

duced to kilowatts by further multiplication by 746 and dividing the result by 1000.

Output

Although the output of a generator can be calculated from the ammeter and voltmeter readings (and in the case of alternating-current work, power factor meter readings) more accurate results may be had by the employment of properly calibrated wattmeters and under variable load conditions, watthour meters.

Voltmeter, ammeter and wattmeter (or watthour meter) connections for output measurements of a two-wire direct-current generator are shown in Fig. 4; with properly calibrated instruments, the product of the instantaneous readings of the voltmeter and the ammeter should correspond with that of the wattmeter.

Connections of instruments used to measure the output of single phase alternating-current generators are similar to those used for two-wire direct-current work, but due to the characteristics of alternating current apparatus, the output of the machine as indicated by the volt and ammeter readings is only the same as that indicated by the wattmeter when the power factor of the circuit supplied is at unity or 1. In order, therefore, to secure the actual number of watts delivered by a single-phase alternating current generator, it is necessary to multiply the product of the volt and ammeter readings by the power factor expressed as a decimal.

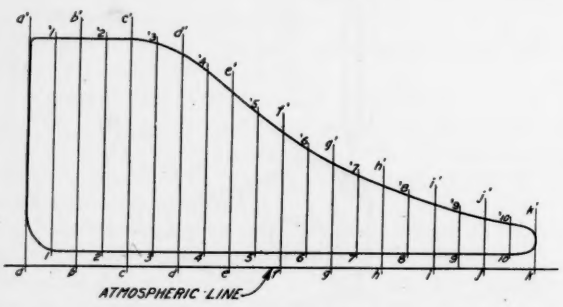


FIG. 3. THE AVERAGE LENGTH OF THE NUMERICALLY DESIGNATED ORDINATES WHEN MULTIPLIED BY THE SCALE OF SPRING USED WILL GIVE THE MEAN EFFECTIVE PRESSURE

Where the machine is of the two-phase type, the output, according to volt and ammeter readings, is equal to twice the product of the number of volts indicated, the number of amperes of current flowing through any one conductor, and the power factor; this, however, under the assumption that the phases are balanced. Under unbalanced conditions, each phase must be treated as a separate single-phase circuit and the results added.

In the case of a three-phase generator supplying current to a balanced load, but one ammeter connected in series with any one of the three leads, and a single voltmeter connected across any two of the lines, are necessary. Multiplying the square root of 3 by the number of amperes flowing through any one line times the voltage across any two of the lines times the power factor will give as a result the output of the machine in watts.

Under conditions existing in a purely non-inductive balanced circuit, the measurement of energy may be by means of a single wattmeter connected as shown in Fig. 5, while on unbalanced systems, either inductive or non-inductive, two single-phase wattmeters connected as in Fig. 6, are necessary. The algebraic sum of the readings gives the total energy delivered.

Field and Armature Losses

By inserting a low-reading ammeter in the shunt circuit and connecting a voltmeter across the terminals of that circuit in the manner indicated in Fig. 7, the shunt field excitation loss expressed in watts may be obtained from the product of the instantaneous volt and ammeter readings. About 20 per cent of this loss will be in the field rheostat.

The series excitation loss is equal to the product of the total current delivered by the machine and the combined parallel resistance of the series field coils and the series shunt.

If the resistance of this is not known, it may be quite readily obtained by employing a Wheatstone bridge, or

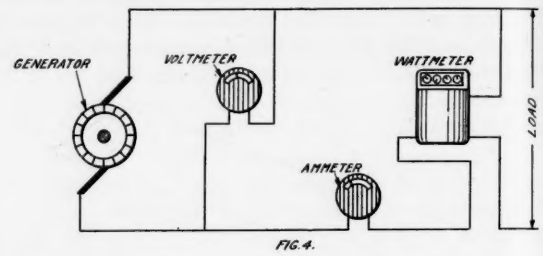


FIG. 4.

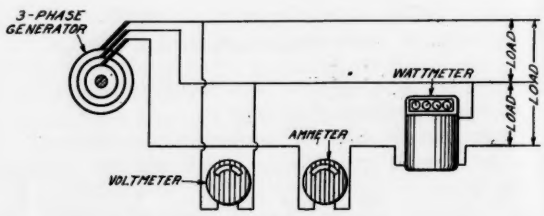


FIG. 5

FIG. 4. INSTRUMENT CONNECTIONS USED FOR CHECKING OUTPUT OF DIRECT CURRENT AND SINGLE PHASE ALTERNATING CURRENT GENERATORS

FIG. 5. CONNECTIONS EMPLOYED TO MEASURE OUTPUT OF 3-PHASE GENERATORS SERVING BALANCED, NON-INDUCTIVE LOADS

if such an instrument is not available, the voltmeter-ammeter method will prove satisfactory. Use the scheme of connections indicated in Fig. 8, where V is the voltmeter, A the ammeter, SF and SS the series field and series shunt respectively under test, B a low-voltage battery, and CS the control switch. Upon closing the switch, a flow of current is established, the rate of which is as indicated by the ammeter A, while the reading of voltmeter V indicates the voltage applied across the terminals of the series field and its shunt. The resistance, in ohms, of the circuit under test will then be equal to the quotient obtained by dividing the voltmeter reading by that of the ammeter.

To determine the copper losses occurring in an arma-

ture, it is necessary to determine, first, the resistance of the armature winding, which in the case of a two-pole machine armature may be done by the volt-ammeter method with the scheme of connections shown at A, Fig. 9. With a low-voltage current flowing through the armature, note the reading of the instruments and calculate the resistance of the armature winding by dividing the number of volts indicated by the reading of the ammeter.

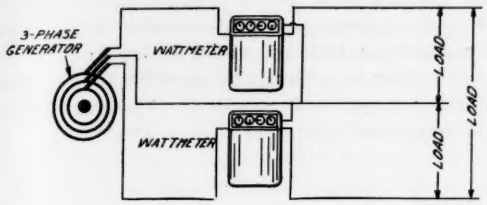


FIG. 6. CONNECTIONS EMPLOYED TO MEASURE OUTPUT OF 3-PHASE GENERATORS SERVING UNBALANCED, INDUCTIVE LOADS

In the case of a four-pole machine, use the scheme of connections indicated at B, Fig. 9, and where a six-pole machine armature is under test, employ the connections shown at C, care being taken, however, to see that the test contacts are properly placed. Where the armature is that of a two-pole machine, place the test contacts diametrically opposite to one another, while in the case of a four-pole generator armature, these will have to be placed on the quarter; in a like manner, six and eight-pole machine armatures will require placing the test leads 60 and 45 deg. apart respectively.

Actual contact of the instrument leads may be made

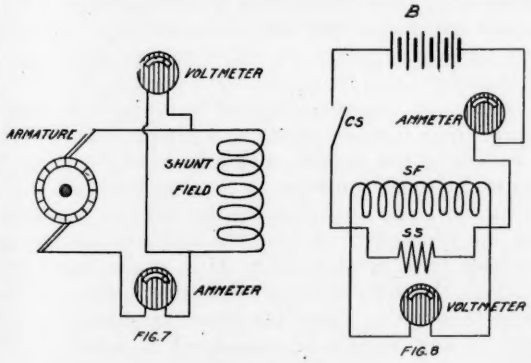


FIG. 7. CONNECTIONS EMPLOYED IN DETERMINING SHUNT FIELD LOSSES

FIG. 8. INSTRUMENT CONNECTIONS USED TO DETERMINE RESISTANCE OF SERIES FIELD AND SHUNT

by temporarily soldering these to the proper segments. With the instrument leads connected to the brush holders, as is frequently done, accurate results will not be obtainable, due to the voltage drop caused by brush and brush contact resistance, which may be from 1 to 2 v. For all practical purposes, however, brush and brush contact resistances should be taken into consideration.

Knowing the value of the resistance of the armature winding, the loss in watts occurring in the armature under any given load may be readily calculated by multiplying this factor by the square of the number of amperes of current flowing.

Friction Losses

For moderate speed bearings with ring lubrication and light machine oil, the friction loss for each bearing may be approximately determined by multiplying the product of 0.8 of the bearing diameter in inches and the bearing length in inches by the result obtained in raising to the two-thirds power the quotient of the rubbing velocity in feet per minute divided by 100.

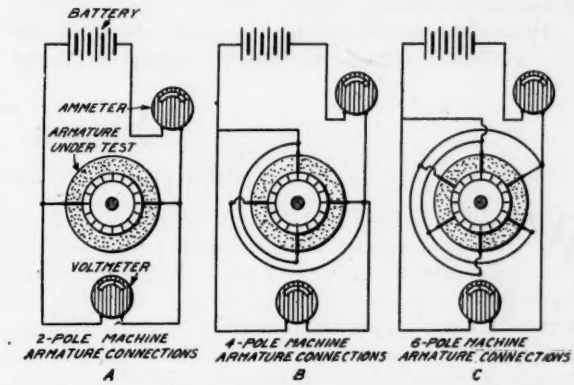


FIG. 9. SCHEME OF CONNECTIONS EMPLOYED IN MEASUREMENT OF ARMATURE RESISTANCE

The total losses due to friction in any given generator are obtained by noting the amount of power required to drive the machine idle and with all field connections open to prevent the building up of voltage.

Efficiency and Regulation

In the case of any generator, the input plus the sum of the various losses is equal to the output, and the efficiency of the machine may be expressed as the ratio between the output and the input, which multiplied by 100

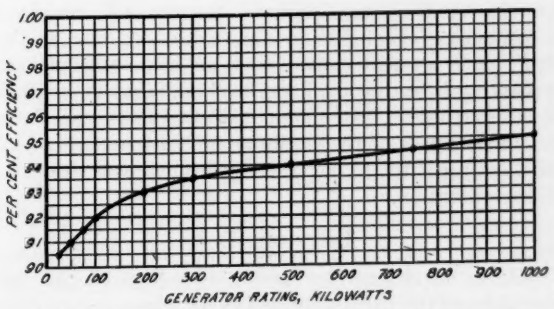


FIG. 10. AVERAGE EFFICIENCIES OF VARIOUS SIZED GENERATORS

will present it on a percentage basis. Where the generator in question is of the alternating-current type, the output must be expressed as the product of the square root of 3 times the voltage across any two lines, times the current flowing through any one line times the power factor expressed as a decimal.

Average efficiencies of generators of various capacities are as shown in Fig. 10. Similar curves for any

machine may be obtained by plotting the efficiencies at various loads, the per cent efficiency for each point being secured in the manner stated above.

Voltage regulation of a generator which is the percentage of the full-load terminal electromotive force that the terminal electromotive force decreases from no load to full load, varies not only with the type of generator but also to some extent according to the inherent characteristics of the individual machine. It may be obtained by dividing the difference between the full-load voltage and the no-load voltage, by the full-load voltage, the voltage readings being taken with the speed of the machine constant.

Assume a machine having a no-load voltage of 110 and a full-load voltage of 100; its regulation would then be equal to the difference between 110 and 100, divided by 100, or 10 per cent.

Except in power service where a considerable drop in voltage results in a heavy current flow, voltage regulation has practically no effect upon the efficiency of a generator.

In going over any generator for the purpose of discovering, reducing and eliminating unnecessary losses, the inspection of the mechanical parts of the machine, such as the bearings, the condition and working of brushes and commutator, and the condition of the various joints and connections, would be the first in order. This should then be followed by tests for the determination of friction and windage losses and those occurring in the various windings.

Steam Turbines

INSPECTION AND TESTING. BY A. D. BAILEY

WHILE MANUFACTURERS of steam turbines have been gradually improving the efficiency and reliability of this type of prime mover, additional responsibility has been placed upon the engineers who operate this equipment to maintain this high efficiency. This improvement has been brought about not only by changes in design, but by better workmanship, more careful construction and observance of the most minute details of construction so far as the operating parts of the turbine are concerned, hence the wonderful efficiencies being obtained with the more recent turbines can be maintained only by keeping this equipment in first class condition and by observing all the fine points of operation, as the neglect of seemingly trifling faults, each one contributing a small amount to decrease the general efficiency, may cause a net decrease which will seriously affect the operating efficiency of the machine. The maintaining of pristine efficiency is distinctly up to the operating engineer.

In order to know at all times the condition of the prime mover it is essential that some means be provided for measuring the input to the turbine as well as the output, and we find today, that all large stations are supplied with testing equipment of one kind or another for this purpose. Instrument manufacturers have been alert to the needs of the engineer and instruments of all kinds are plentiful to cover every branch and detail of turbine operation.

INSTRUMENTS

WHILE apparatus for actually weighing the condensate from the turbine is, of course, the most accurate and satisfactory means of checking turbine performance, the installation of such equipment is necessarily expensive. Manufacturers of steam flowmeters and water measuring devices have improved their equipment to a point where it is reliable and accurate so that by this means the efficiency of the prime mover may be checked continuously and any slight variation noted and investigated. The error of these measuring devices should be no greater than that of the standard electrical instruments used for measuring the generated output, and comparative readings under similar conditions should give comparative results.

Pressure gages, thermometers and all other instruments used in connection with the operation of the turbine should be not only correct, but should be periodically and frequently checked to maintain their accuracy, as the readings of these instruments are absolutely essential in checking turbine performance. It should be borne in mind, however, that while pressure readings taken in the various stages of the turbine will immediately indicate any serious defective condition of the nozzles or blading, these instruments are not sufficient to indicate small defects which may have a decided bearing on the general efficiency of the machine.

Placing of these instruments is also a matter of considerable importance and it is necessary that they be so placed as to give a true record of the conditions; also that the record will show conditions at vital points. In this connection, cognizance should be given to the steam pressure drop through valves and piping leading from the boilers to the turbine and care must be taken to measure the initial pressure at the throttle valve or admission valves as provided in the builders' contract. Since the pressure drop from the boiler drums may in some cases be as much as 15 or 20 lb. under heavy load conditions, this item must be taken into account, as it is a variable quantity depending upon the steam flow and cannot in any manner be considered as a part of the turbine performance. The installation of a strainer in the steam line, while imperative, may introduce a considerable reduction in steam pressure, due to the restriction of area of the steam line, particularly if it becomes partially clogged over a considerable period of time.

In locating instruments which measure the temperature and pressure of exhaust from the turbine, the same care must be taken, as the manufacturers' guarantee is generally based on conditions at the exhaust flange of the machine.

Since with condensing equipment the pressures at this point are below atmosphere, it is particularly necessary that all connections to instruments be tight and be made in such a way that the velocity of the exhaust

steam will not in any way affect the reading of the instrument. The instruments must, however, be so connected as to give average conditions throughout the area of the exhaust opening.

Condenser

The condenser should be so constructed that there will be a minimum pressure drop between the exhaust outlet of the turbine and the exit to the vacuum pump, and the condensate, as fast as it is formed, must be taken care of so as to reduce the refrigerating effect of the lower portion of the condenser; temperature of the condensate must be maintained as near as possible to that of the vacuum at the turbine exit. In some condensers the condensate is returned through a bank of tubes placed directly in the exhaust from the turbine so that it will be heated to the temperature of the exhaust steam.

Air leaks in the condenser and low-pressure casing of the turbine not only place added work on the air pump, but may result in the formation of air pockets in the condenser which seriously impair the efficiency of the condensing surface, while leakage around the tube packing allows the fouling of the condensate with impure condensing water which directly affects the cleanliness of the boilers and may, in addition, seriously affect the performance of condenser and auxiliary apparatus.

If the turbine is provided with an atmospheric relief valve, this must also be checked as it is a potential source of air leakage, while the water from the seal of this valve which is generally from the same source as the condensing water, may be drawn into the condenser at this location as well as through the tube packing.

Testing Input

In order to check the input of the turbine properly, it is necessary that the steam pressure and temperature be known at all times, and that a steam flowmeter be installed in the line near the throttle, or that a water measuring device be installed giving a record of the amount condensed from the turbine. Builders' specifications generally provide that steam pressure and temperature shall be measured at the throttle valve on the header side, or if the manufacturer provides a strainer which is installed outside of the throttle valve pressures and temperatures should be taken at the inlet to this strainer. While indicating instruments are sometimes considered sufficient at this location, it is advisable to install recording instruments as well, particularly in the case of large units, and it will be found that practically all instrument manufacturers have standard recording instruments for this purpose. The pressure in the initial stages of the turbine should also be known, although an indicating gage is probably sufficient for this purpose. So far as the exhaust pressure is concerned, it is well to install a long bulb low-reading thermometer in the exhaust nozzle of the turbine just above the exhaust flange, and as near as possible to this connect a standard mercury column for measuring vacuum. The connection to the latter instrument should be carried inside the exhaust casing and should be perforated or constructed in such a manner that the steam velocities will not affect the reading of the instrument. As this reading is one of the most important in connection with turbine performance, the greatest care should be exercised to connect it properly and to maintain it in perfect condition so that reliable readings will be obtained. While there are a number of spring-operated vacuum gages on the market, both indicating and recording, a mercury column instrument should always be at hand for comparison, and the reading of this instrument compared with that of a mercury column barometer. By a comparison of the temperature and pressure readings in the exhaust nozzle, leaks in the low-pressure casing of the turbine or in the low-pressure shaft packing, may be readily detected.

Other readings which have a decided bearing upon

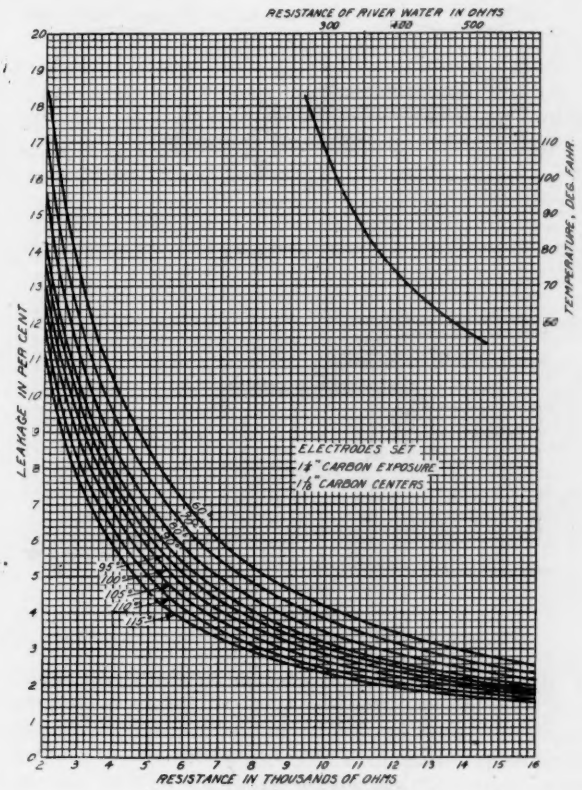


FIG. 1. CONDENSATE LEAKAGE-RESISTANCE CURVES

turbine performance are the temperatures of the circulating water entering the condenser and discharging from it, as well as the temperature of the condensate leaving the condenser; also the pressure in the exit to the dry vacuum pump.

In order to get full benefit of these instruments, it is necessary that readings be taken at regular and frequent intervals and that a record be made of the average pressures and temperatures for each day, along with the total input and output of the turbine. These daily averages should be entered and from them monthly averages deduced, which may later be used for comparison, and which form a ready means for checking relative turbine performance at any time.

Attendants and operators should be taught the meaning and importance of the readings of these various instruments and should be supplied with forms for properly recording these readings. Any changes in conditions can be noted by them and reported, and the cause of the trouble investigated and removed whenever it is purely operating condition.

Condensate Measurements

Where water measuring instruments for measuring the condensate are used, another instrument is necessary for the measurement of leakage in the condenser, and since this is included with the condensate in any water measuring device, this amount must be deducted in determining the efficiency of the turbine. It has been

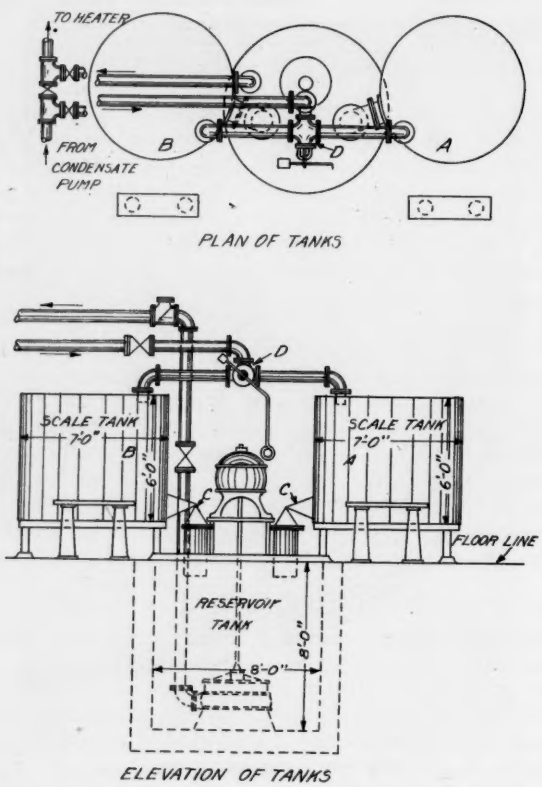


FIG. 2. TANK LAYOUT FOR WATER RATE TESTS ON TURBINES

customary to make a leakage test whenever the unit was taken out of service, by measuring the discharge from the condenser after the steam has been shut off and all priming lines to the condenser have been closed. This is not satisfactory, however, as it has been found that condenser leakage varies with the load and that the leakage shown under no load conditions is not in any way indicative of the leakage under full load conditions. It is therefore essential that this leakage be measured while the turbine is in operation, and the instrument to be described does this satisfactorily as it takes into account all water entering the condenser from an external source, either through leaks, priming valves or through make-up for the boiler feed.

This is known as the electrical resistance method and the apparatus consists of two properly insulated electrodes suitably set in an immersing jar which holds a sample of the water of condensation to be tested, a voltmeter measuring burette and connections from a 220-v. alternating current circuit. The resistance of a number of mixtures of pure condensate and condensing water in various known proportions can be determined experimentally for various temperatures, and the results drawn in the form of a curve giving percentage of condensing water per unit of resistance. Having established these curves for the instrument, the percentage of leakage may be determined at any time while the machine is in operation by drawing off a sample of the condensate and checking its resistance.

Instruments of this type have been installed permanently in the condensate discharge from condensers and an indicating dial perfected for showing the leakage at all times.

Apparatus for actually weighing the condensate from the turbine is of course the most accurate and satisfactory means of checking turbine performance and the installation of such equipment affords a means for periodic tests, the reliability of which cannot be questioned. Figure 2 shows the arrangement of piping, tanks and scales for such a water-weighing installation. The outfit consists of two tanks on individual scales, each having a large, quick-opening valve near the bottom and so arranged that they can be drained into a large reservoir below. A return pump removes the water from this reservoir and returns it to the feedwater heater or to some other part of the system from which it came. The sketch also shows the piping arrangement with valves for diverting the condensate from the condensate pump to the weighing tanks. Three-way valve "D" is so arranged that water may be discharged into either tank A or B so that one tank of water may be weighed and discharged into the common reservoir while the other tank is being filled. The weighing scales have double beams so that one will take care of the empty tank weights and the other will show directly the weight of the water.

Test Corrections

Wherever possible tests should be conducted with the turbine operating as nearly as possible to the builders' specified conditions regarding steam pressure, temperature and vacuum and any variation from these specified conditions should be noted and corrections made for them according to the builders' specified corrections. The following list of corrections is approximate:

1 per cent to be added to the specified water rate for each 10-lb. variation in steam pressure below that specified.

1 per cent each 12 to 15 deg. F. variation in the steam temperature below that specified.

6 per cent for each 1 in. of mercury variation in absolute condenser pressure below that specified.

A corresponding reduction in water rate to be made for corresponding increases in temperatures and pressures. In this case also it is necessary to determine condenser leakage.

Inspections

If the efficiencies obtained by water rate tests at any time disagree seriously with the builders' guaranteed efficiencies, the turbine should be taken out of service at the first opportunity and a thorough inspection made of the valves, nozzles and blading to determine if their condition is in any way changed from the original. While inspections of the smaller parts of the turbine should be made regularly at frequent intervals, the machine as a whole should be thoroughly inspected and overhauled each year; from throttle valve to exhaust casing each part should be examined most carefully and all defective or worn parts renewed or repaired.

The throttle valve must be tight, as otherwise the leakage of steam into the turbine while it is shut down, may cause unequal heating of the shaft and wheels, resulting in a distorted shaft and warped wheels, so that considerable difficulty may be experienced in starting, due to vibration. Repeated unequal expansion and contraction of the various turbine parts may cause permanent sets which will seriously affect the operation of the machine. If a strainer is installed in the steam line this should also be inspected to make sure that the screen and its supporting framework are intact and to remove any foreign objects which may be lodged against the screen.

In case of a multivalve governor control, the proper seating of these valves is essential to economy as steam which is wire-drawn through leaking admission valves is useless in the turbine and on account of its reduced pressure and consequent velocity, is a dead loss.

The governor should be given particular attention to insure its perfect operation and after repairs are made its range should be checked so that it will satisfactorily control the turbine from no load to full load under all conditions. The emergency or runaway governor should be tested each time the unit is taken out of service—at least once each week—and should be examined for wear and its operation carefully checked before the machine is again put into service.

If the turbine is equipped with overload valves admitting steam to advance stages of the turbine under heavy load conditions, these should be likewise inspected and any leakage corrected. The most favorable load at which these should operate should be determined by test and both the load and relative stage pressures noted so that the operations of these valves may be readily checked from time to time.

High- and low-pressure shaft packing should be dismantled and, if necessary, refitted. The latter, particularly, must be kept tight to prevent air leakage into the exhaust casing with consequent reduction of vacuum and efficiency. The leakage of the former, while not so important, is unsightly and at times dangerous, besides being the cause of wastage of steam. The packing on the dummy rings of machines of the types employing these parts are also due for inspection, and clearances should be carefully taken to get the proper results with the machine in operation.

The main bearings should be removed and the condition of both shaft and bearings carefully noted. The clearances specified by the manufacturer should be maintained and the position of the shaft, axially and radially, carefully checked. On vertical machines the manufacturers generally allow approximately 0.001 in. clearance per inch of shaft diameter, while with horizontal machines this may be slightly increased.

The oiling system should be completely cleaned, the oil tested and any signs of the shaft pitting due to acidity of the oil or electrolysis carefully investigated and proper steps taken to stop the trouble. All oil passages and grooves should be carefully cleaned and any carbonized oil or sludge removed. The thrust bearings should also be included in the bearing inspection.

If the bearings must be renewed, it is wise to follow the manufacturer's recommendations regarding the mixture used for bearing metal, as the failure of a bearing while the machine is in service would be disastrous.

Nozzles and Blading

It is advisable to run tests to determine the loss in efficiency due to eroded or faulty blading or nozzles and the point determined at which renewals are justified.

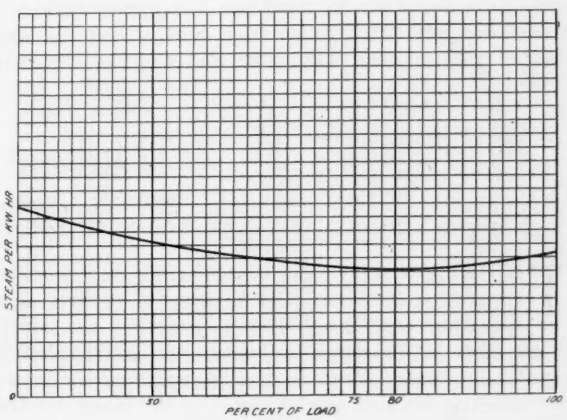


FIG. 3. TYPICAL TURBINE WATER RATE CURVE

There are, however, many vital points which might be overlooked except by an experienced turbine man, such as the fastening of blading and space blocks, the proper locking of cap bolts and rivets used in the construction of the turbine parts, the cracking of nozzles, wheel cracks, proper fastening of shroud bands and other similar items which are matters of the greatest importance, as their neglect might not only jeopardize the turbine itself but might endanger the lives of the attendants.

To maintain the efficiency of the turbine all dents in nozzles and blades should be straightened and the parts restored to their original shape. Any dirt or deposit on the blades or between them should be removed and the rough edges, due to erosion or water wear carefully straightened and sharpened. Alinement of nozzles and blades should be checked and the wheels and shaft so adjusted as to give proper clearance and to insure proper operating conditions. With impulse turbines the important clearance is that between nozzles and wheel blades, while with reaction turbines uniform peripheral clearance of the revolving and stationary

blades is the important thing. If it is found necessary, due to wear or damage, to renew any of the blading, this work should be done only by an experienced man, as the future operation of the turbine will depend on the quality of this work.

The casing should be carefully inspected and proper care exercised to insure tight joints throughout, as leaks on the high-pressure casing are unsightly, wasteful and dangerous, while leaks on the low-pressure end are detrimental to the best vacuum conditions. Bolts, particularly those subjected to high temperatures, should be renewed or the old ones should be annealed before being replaced. After the machine is assembled all low-pressure casing joints subject to vacuum should be tested for leaks and painted with heavy paint which will be drawn into the interstices and cracks by the vacuum. Bolt heads and nuts on this part of the machine should be similarly painted where they face against the turbine casing.

Before closing up the machine, the greatest care should be taken to clear out all loose articles of every description. Many cases have occurred where small tools have been left in machines by careless workmen and considerable damage has been done, so that the unit was in poorer condition after the overhauling than before.

Covering on the high-pressure casing should be replaced to prevent radiation and to reduce condensation in the earlier stages of the machine. Variations in the temperature of the casing are liable to cause uneven expansion and contraction, possibly resulting in cracks, and leaky joints.

Before starting, the machine should be gradually warmed before it is turned over and even after it is in motion it should be brought to speed very gradually and the range of the main governor, as well as the emergency governor checked before it is put into service. The load should be increased gradually to allow gradual increase in temperature throughout the turbine.

While it is good policy to conduct tests immediately before and after overhauling a machine of this kind in order to determine the benefits of the overhauling, the importance of the annual inspection should not be overlooked in case the benefits are not readily apparent.

A record should be kept of each overhauling, reporting in detail all work done and conditions as found, as well as the condition of the machine at the time it is again put into service. Special forms may be devised for easily and quickly making such a record, listing inspections, clearances, fits, etc. In this way the history of the mechanical condition of the machine is always available for reference.

Turbine Operation

INSPECTION, OPERATING CONDITIONS AND CHECKING OPERATION. BY J. WILSON

TO OBTAIN THE best results from any piece of apparatus, it is essential to keep it in first class operating condition. This means that the equipment must be thoroughly cleaned and overhauled at regular intervals, therefore when considering the means for insuring the best performance of the steam turbine it is advisable to consider first these periodic inspections, as incorrect adjustments and rough operation will result in reduced efficiency.

In making a turbine inspection the following should receive careful attention:

The steam strainer should be thoroughly cleaned and, if it is in bad condition, should be renewed. A dirty strainer prevents the free passage of the steam, which causes an excessive drop in pressure through the strainer, and a broken one might permit foreign matter such as pieces of gasket, etc., from the steam pipes, to pass into the cylinder. This foreign matter might lodge in the blading, obstructing the openings through the blades and by reducing the inlet area, cause the bypass valve to open earlier or at a lower load than usual. As the efficiency of the unit is reduced when operating on the bypass, the steam consumption of the turbine would be increased unnecessarily.

CHOKING OF BLADES AND CLEANING THEM

IN SOME plants where it is difficult or impossible to obtain clean feed water, mud and other impurities are carried over from the boilers to the turbine with the steam. These impurities sometimes adhere to the blades, choking up the openings between them. In such cases, the blading should be thoroughly cleaned, otherwise loss in economy will be incurred due to the bypass valve opening at reduced load as already mentioned.

Choking up of the blades, in addition to reducing the economy and capacity of the unit, is dangerous, as it will unbalance the spindle in an axial direction, thereby throwing additional load on the thrust bearing. If the additional load should be sufficient to cause excessive wear on the thrust bearing rings, serious trouble will result, as the dummy pistons will come in contact with the dummy packing rings. When a heavy deposit is found in the blading, it is advisable in addition to cleaning blades thoroughly to give the condition of the feed water careful attention.

Method of cleaning the blading depends on the nature of the deposit. In some cases a spray of compressed air and kerosene is sufficient to remove it, whereas in other cases the deposit adheres to the blades so tenaciously that it is necessary to scrape the blades. In the latter case care should be taken to see that the blades are not scored, or otherwise injured in the process. Another method which has proved successful in practice is to connect a small hand pump to the steam pipe leading to the turbine and to pump small quantities of kerosene into the steam at short intervals while the unit is in operation. The kerosene has the effect of loosening the deposit slowly, allowing it to be carried through the turbine to the exhaust. A barrel or two of kerosene injected into the steam line as above has been known to clean the blading satisfactorily, but as already stated,

success or failure depends entirely upon the nature of the deposit.

It should be fully understood that it is not a common occurrence to find a deposit in the blading and the foregoing suggestions have been made for the benefit of the few cases where such conditions occur.

Air Leaks

Spindle glands at both ends of the cylinder should receive careful attention, for a defective gland which allows air to leak into the exhaust space reduces the economy of the unit. Loss of 1 in. of vacuum means, in general, an increase in the steam consumption of from 5 to 10 per cent, therefore inleakage of air at these points should be carefully guarded against.

Water-seal glands provided by several manufacturers of steam turbines are probably the most satisfactory and give the least trouble provided the water supplied for sealing purposes is free from acid and scale-forming impurities. If on examination, scale is found in the runners and casings, it should be carefully removed as this reduces the efficiency of the runner and might allow air inleakage. Also if the scale is excessive, the clearances between the rotating and stationary elements may be reduced until contact occurs when, owing to the high speed of rotation, serious trouble will result.

Condensate from a surface condenser is excellent for sealing purposes, also the drips from steam lines, heating coils, etc. If suitable water is difficult or impossible to obtain from available sources, a simple water purification system can be easily and cheaply installed. Temperature of the water to the glands should not exceed 100 deg. F., as the churning of the runners heats the water and with warm inlet water might cause vaporizing which would be objectionable, as the vapor escaping from the casings would condense on the bearing housings and this water might find its way into the oil.

Bearings

Main bearings should receive attention. If of the spherical type the bearings should fit snugly in the ball seats without any measurable clearance, but free to move to take up the correct position to suit the deflection of the shaft. In other words, they should be a close working fit as any clearance will tend to cause vibration. If it should be necessary to add shims, care must be taken to maintain the spindle central in the cylinder.

It is not usual to find wear in the bearings, but if wear has taken place, the bearing should be taken up and the clearance between the journal and the bearing reduced to the original amount. The clearance in a turbine bearing is in general from one to one and one-half thousandths of an inch per inch of diameter. This is sufficient to allow the shaft to float on the oil film without unnecessary clearance to cause vibration. Oil grooves in a turbine bearing are sometimes considered by operators unnecessarily large. They should not be decreased, however, without the approval of the manufacturer as in many designs of turbines the oil is the medium for removing the heat generated in the bearing by the high speed of rotation. As the smallest interruption in the oil supply to the bearings would have disastrous results, care should be taken to see that all oil grooves and oil pipes are clean and unobstructed.

Oil Cooler

Frequently the oil cooler is the piece of apparatus most neglected on a steam turbine. This should be overhauled in the annual inspection just as carefully as any other part of the apparatus. All mud and foreign matter should be removed from the water box and tubes and, if the inside of the tubes has become coated with a gummy deposit, this should be washed out with hot water, and if necessary, with lye. After cleaning, care should be taken to rinse the tubes several times with hot water to remove all traces of the lye.

Sometimes it happens that a thin hard scale deposits on the outside of the cooling tubes which is difficult to remove. Washing is useless and scraping, owing to the close proximity of other tubes, is impossible. In such cases a home-made sand blast has proved very effective, removing the scale in a shell like formation, not only from the easily accessible surfaces, but also from the tubes inaccessible by other means.

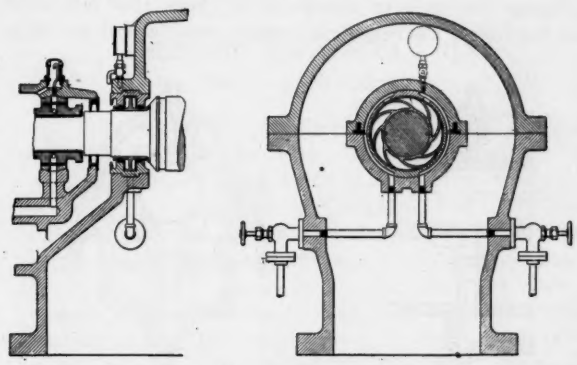


FIG. 1. SECTION OF WATER SEALED SPINDLE GLAND

If necessary, the whole oil system should be thoroughly washed out and new oil put in. On some machines oil filters are installed to filter the oil automatically. This operation is usually on the partial filtration principle, drawing off a certain proportion of the dirty oil continuously at the lowest point in the oil system, filtering it and returning the cleaned oil to the system. This is an excellent device, but the expense of the equipment frequently prohibits its use in connection with small installations. In such cases it is probably best to drain off all the oil at stated intervals, filling up the tank with new oil and using the old on station auxiliary apparatus.

Valves and Packing

Inlet and bypass valves and seats should be examined and re-ground if necessary. If the unit is operated for long intervals at a fairly constant load, it sometimes happens that scale or rust forms in the balance cylinders immediately above the valves. This should be removed as any increase in the load, creating an additional demand for steam, will cause the valve to open wider and the valve packing rings to stick on the scale in the balance cylinder thereby upsetting the governing of the

unit. Packing rings should be examined to see that they retain their life and have sufficient spring to prevent steam leakage. If they have been subjected to superheated steam, the opening between the ends of the rings should be checked to see that the clearance is maintained, as superheat causes cast iron to grow and this growth might eliminate all the original clearance and cause the rings to stick.

There is little spindle end thrust in turbines of modern design, therefore, unless the balance is affected by outside influences such as foreign matter choking the blading, it is to be expected that the turbine thrust bearing will be in first class condition. As a matter of fact, it has frequently been noted that after years of operation the tool marks still show on the thrust rings. The thrust bearing, however, should be thoroughly cleaned and the collars re-ground, if they have become scored or otherwise damaged.

The chief function of the thrust bearing is to maintain the desired minimum clearance between the rotating dummy rings on the spindle and the dummy strips mounted in the cylinder. It is advisable before again placing the unit in operation to check this clearance, as too fine a clearance may cause trouble and too large

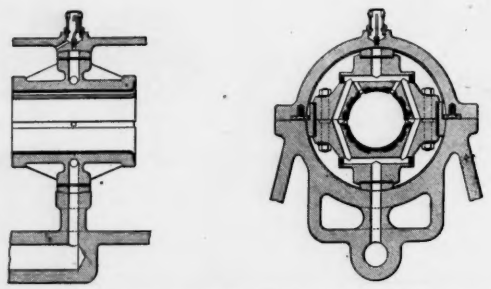


FIG. 2. BALL AND SOCKET TYPE OF BEARING SHOWING MEANS OF OIL SUPPLY

a clearance means excessive steam leakage over the dummies and consequent decrease in the steam economy of the unit. Clearances should be checked when the machine is heated up as nearly as possible to the normal operating temperature. In general, the clearance will vary from 0.010 to 0.025 in., but as the permissible minimum clearance varies on units of different size and type, it is advisable to obtain the correct clearance from the manufacturer of the equipment.

Foregoing are the principal features requiring special attention to insure satisfactory and economic operation; but of course in addition many small details on the equipment such as the regulating governor, safety stop governor, direct connected oil pump, auxiliary oil pump, gears, etc., should not be overlooked in the annual inspection as the inaccurate functioning of any of these parts may cause considerable annoyance and anxiety and the failure of any one of them may result in a shut-down, probably at a most inconvenient time.

Operating Conditions

To obtain the greatest economy a machine should be operated under the steam and vacuum conditions specified in the contract. To operate with a lower vacuum than the machine was designed for incurs greater loss in economy than the difference in the available B.t.u.'s indicates, as the volume and consequently the velocity of the steam through the exhaust blading is decreased and instead of the steam doing work in the last few rows of blading, the blades may be retarded and act to pump the steam into the exhaust.

Similarly, to operate with a much better vacuum than the machine was designed for shows little improvement in economy, as the area through the exhaust blades is not sufficient to take advantage of the greatly increased volume of the steam.

To increase the initial steam pressure, provided the machine will withstand such, will slightly improve the economy, and a moderate amount of superheat will prove beneficial by removing the moisture from the steam; but the question of the adoption of superheat should be approached with caution. The volume of superheated steam is greater than that of saturated steam at the same pressure and as the blading can pass only a definite volume of steam in a given time, it follows that less weight of superheated steam can pass through the inlet blades in a given time; consequently, when operating with superheat the bypass valve will open earlier than it would if the unit was operated with saturated steam, and as the efficiency of the unit is reduced when operating on the bypass it can be shown in some cases where the regular load on the unit with saturated steam is just under the point where the bypass valve will open, that to operate with superheated steam would be detrimental.

Again, superheated steam has the effect of causing cast iron to grow, consequently, if the valves are of cast iron, the growth of the valve and valve rings might cause serious difficulties with the governing of the unit, and in extreme cases the steam chest and cylinder might be affected.

The most economical point to operate a turbine is of course at the highest load obtainable without the aid of the bypass valve. The reason for this is readily understood as the friction load of the machine within reasonably small limits remains the same whatever the load on the unit may be, consequently the greater the load the smaller the percentage of loss due to friction. As soon as the bypass opens, the efficiency falls off as the steam entering through the bypass valve, acting on fewer rows of blades, gives up less energy, consequently more steam is required to produce a unit of power. It can be readily seen from the above that the bypass valve should be adjusted so that it does not come into operation until the maximum load without the bypass has been reached and the addition of any more load would cause the unit to slow down. This point is usually reached when the inlet pressure approaches within about 5 lb. of the steam pressure immediately ahead of the throttle valve. The bypass valve can be readily adjusted to open at the desired time by lengthening or shortening the connecting rod or the bypass valve spindle.

Checking Economy

There is no simple and ready method of accurately checking the performance of a steam turbine to deter-

mine whether or not it is fulfilling the contract guarantees, or of comparing the performance to see that the economy is maintained. To measure the steam used in the unit, the water must be weighed either before it enters the boilers or after it is condensed in the condenser. This necessitates tanks and scales of considerable capacity. When the test is made by weighing the condensate, a meter must be inserted in the water line to the glands so that proper corrections can be made for the water leaking from the glands into the exhaust. A calorimeter must be installed in the steam line to the turbine to determine the amount of moisture in the steam and thermometers must be furnished to check the temperature of the steam at the steam chest.

Gages are, of course, necessary to obtain the throttle and inlet pressures, also the various electrical instruments to determine the load accurately. Gages and electrical instruments should be calibrated both before and after the test, and the test results suitably corrected. During the test, which to insure accuracy should last several hours, a steady load must be maintained.

WATER RATES OF TURBO-GENERATORS

(60-cycle, 2300-volt a.c. generators)

Condensing				Non-condensing	
Steam at 150 lb. gage, 28-in. vacuum		Steam at 200 lb. gage, 150 deg. superheat, 29-in. vacuum		Steam at 150 lb. gage Exhaust to Atmosphere	
Rating kw.	Water rate, lb. per hr. per kw. incl. Excitation	Rating kw.	Water rate, lb. per hr. per kw. incl. Excitation	Rating kw.	Water rate, lb. per hr. per kw.
50	32.0 to 42.0	4,000	12.0 to 13.3	50	52.0 to 60
100	21.2 to 27.5	5,000	11.9 to 13.0	100	43.5 to 50
200	19.5 to 25.0	7,500	11.6 to 12.6	200	38.4 to 44
300	18.7 to 22.5	10,000	11.2 to 12.3	300	36.5 to 42
400	18.2 to 21.0	12,500	11.1 to 12.0	400	35.2 to 40
500	17.7 to 19.3	15,000	11.1 to 11.8	500	34.1 to 38
600	17.4 to 19.0	17,500	11.0 to 11.6	600	33.4 to 37
750	17.0 to 18.8	20,000	11.0 to 11.4	750	32.7 to 36
1,000	16.5 to 18.6	25,000	10.9 to 11.3	1,000	31.7 to 35
1,250	16.2 to 18.3	30,000	10.8 to 11.2	1,250	31.0 to 34
1,500	16.0 to 18.0	35,000	10.7 to 11.1	1,500	30.7 to 33
2,000	15.7 to 17.7	40,000	10.6 to 11.1	2,000	30.1 to 32
2,500	15.5 to 17.4	50,000	10.5 to 11.0	2,500	29.4 to 31
3,000	15.4 to 17.1	60,000	10.5 to 10.9	3,000	28.9 to 30
3,500	15.3 to 16.9	75,000	10.4 to 10.8	3,500	28.6 to 29

*Speeds: 50 to 3500 kw., inc., 3600 r.p.m.; 4000 to 20,000 kw., inc., 1800 r.p.m.; 25,000 kw. and over, 1200 r.p.m.

If the test is to be made by weighing the water delivered to the boilers, a battery of boilers sufficient to furnish steam for the test must be segregated from the rest of the equipment and these boilers used for the purpose of the test alone, all the auxiliaries being operated with steam from some other source. Tests to determine the boiler leakage must be conducted and all condensed steam from drains and drips must be measured, and the amounts credited to the turbine in the final summing up.

When the test is made by weighing the condensate, the condenser must be tested for leakage both before and after the test on the turbine and the results corrected accordingly.

It is frequently impossible to make a test with the steam pressure, steam temperature and vacuum specified in the contract. In such cases it is not sufficient to correct the test results on the basis of the available B.t.u. in the steam under the test conditions and contract conditions, as a turbine built to suit a certain set of conditions cannot operate as efficiently under another set of conditions for the blading is not proportioned correctly for the new condition. It is, therefore, necessary to obtain from the manufacturer correction factors to be applied to the results obtained on test.

A steam flowmeter in the steam line to the turbine will afford a rough and ready means of comparing the performance from day to day, but careful judgment must be exercised or considerable worry and unnecessary investigation may be caused by not giving proper consideration to the effects of varying conditions and fluctuating loads.

A rough comparison of the performance of the machine at stated intervals can also be obtained by means of the station logs. If, for instance, at a given load, with steam and vacuum conditions the same, the inlet pressure should vary appreciably, it is reasonable to assume that some change has taken place in the apparatus or the adjustments and an investigation is warranted. As a matter of fact, if the apparatus is kept in first class operating condition, there is actually little necessity for watching closely the economy of a reaction steam turbine for, owing to the fact that there is no appreciable wear in the blading, sustained economy can be expected after years of operation.

Bending Cast-Iron Pipes

By Mark Meredith

A SUCCESSFUL bending of large straight cast-iron pipes was recently effected in making the line of cast-iron pipe which it had been decided to lay to convey water from the Guayabo River to the town of Present, Cuba. By mistake, no sleeves or curves had been ordered. The canon through which the pipe passes for a distance of about two miles from the dam is crooked, making impossible such easy curves as would be made in the pipe joints. As it might have taken several months to secure additional special pipes, the local engineer decided to bend some of the straight cast-iron pipes. The Cuban workers had frequently bent steel or wrought iron pipes at their sugar mills and they followed the same course of procedure with the cast pipes, with entire success, as they did not break or spoil a single pipe. The pipes were bent to various radii, the shortest being 50 ft. A cradle of old rails was first constructed with the desired amount of curvature, and a fire of hardwood was built under and around the pipes. Six or eight pipes were bent at a time. In one and a half or two hours after starting the fires the pipes were hot enough to bend and settle under their own weight on to the cradle prepared for them. The pipes were 10 in. in diameter with 9/16 in. thickness of shell, their weight being about 760 lb. per 12 ft. length.

Announcement is made that further important changes are to be made to increase the power resources of Turners Falls Power & Electric Co. by raising the entire canal wall and the embankments along the forebay to a height of 2 ft. This plan will greatly increase the amount of water delivered to the wheels from the Connecticut River and enlarge the capacity of Montague City power-house in proportion.

Shafting Losses

From Bearings, Poor Alinement, Tight Belts, or Bad Lubrication. Reasonable Standards and How to Work Towards Them. By H. C. H. Mehciz

BEARINGS, PULLEYS, shafting and clutches of a plant, although among the prominent factors in its efficiency, are often neglected until they reach the stage where their condition absolutely compels attention.

A plant will not only be saved many shut-downs and repair bills, but will improve in efficiency by regular and proper inspections of these parts, and if there be any part not in proper working condition, it can be repaired before any damage is done.

Causes of Power Loss

Causes of excessive power loss in ordinary line shafting, can be directly traced to one or more of four reasons: overloading of bearings, poor alinement, bent shafting and poor lubrication.

The overloading of bearings may be due to bearings being too short, not enough bearings or overtight belts.

Bearings

Many bearings are decidedly lacking in length, which causes trouble and requires excessive lubrication. The pressure per square inch on the projected area of the shaft in the bearing has been found, in good practice, to range from 400 to 800 lb., depending upon the nature of the service to which the bearing is to be put. In general transmission, this has solved itself into certain ratios of length of bearing to the diameter of the shaft. For shafts under 100 r.p.m., 2.75 times the shaft diameter; for 100 to 350 r.p.m., 3 times the diameter.

When not enough bearings are placed along the line shaft, the bearings will be subjected to excessive overloads, which results in undue friction by too much pressure upon the bearings, causing them to heat or to be thrown out of line.

TABLE I. DIAMETER OF SHAFT TO CARRY MAXIMUM ALLOWABLE LOAD MIDWAY BETWEEN BEARINGS AT CENTER OF BAY

Diam. of Head Shaft	DISTANCE BETWEEN BEARING CENTERSA							
	2½ ft.	3 ft.	3½ ft.	4 ft.	5 ft.	6 ft.	8 ft.	10 ft.
in.	in.	in.	in.	in.	in.	in.	in.	in.
2	2	2¼	2⅜	2½	2⅝	2¾	2⅞	3
2½	2½	2⅝	2¾	2⅞	3	3⅛	3⅜	3⅝
3	3	3⅛	3¼	3⅜	3½	3¾	4	4⅛
3½	.....	3½	3⅝	3¾	4	4¼	4½	4¾
4	.....	4	4⅛	4¼	4½	4¾	5⅛	5⅜
4½	.....	.....	4½	4⅝	4⅞	5⅛	5½	5⅞
5	.....	.....	5	5⅛	5⅜	5⅝	6	6⅛
5½	.....	.....	.....	5½	5¾	5	6⅛	6⅞
6	.....	.....	.....	6	6⅜	6⅝	7⅛	7½

The proper distance between bearings of a shaft is governed by the deflection which should never exceed 0.01 in. per foot of length. This, in turn, is dependent upon the diameter of the shaft and the location of the pulleys with reference to the bearings.

For ordinary line shaft work, it is sometimes necessary to place pulleys at various points along the shaft without reference to the bearings. The general rule for line shafts is to place the bearings 8 ft. apart. The table given by Kent will be of value, as it gives the diameter of shaft necessary to carry a load at the center between bearings compared with the diameter for head shafting.

Expressed as a rule, where shafting is used for transmission alone, the distance D, in feet, between bearings, is taken as the cube root of the product of 720 times the diameter, d, in inches, squared ($D = \sqrt[3]{720d^2}$). Where the shaft carries a number of pulleys placed at various points between bearings the distance D′ is equal to the cube root of 140 times the diameter squared ($D' = \sqrt[3]{140d^2}$).

Tight Belts

Overtight belts place undue strain upon the bearings causing excessive friction, heating of bearings and consequent loss of power in shafting. Often belts have been placed on pulleys so tight, that they have caused the breaking of pulleys and bending of shafts, besides wearing out the whole outfit and causing heating and consequent destruction of bearings. Table II, computed by a New England engineer, shows the power required to drive the shafting alone in some average modern cotton mills with first class shafting. These may be taken as a fair showing of the power required in many of the best mills to drive shafting, and it is reasonable to think that not all this power is consumed by a legitimate amount of friction of bearings and belts. Tight belts when there are hundreds or thousands in a mill, easily create great friction on the bearings and would account for the figures.

TABLE II.—HORSEPOWER TO DRIVE SHAFTING

Mill No.	Whole Load HP.	Shafting Alone	
		HP.	Per cent of Whole
1	199	51	25.6
2	472	111.5	23.6
3	486	134	27.5
4	677	190	28.1
5	759	172.6	22.7
6	235	84.8	36.1
7	670	262.9	39.2
8	677	182	26.8

The formula and Table III., as given by Kent, show the normal friction losses for shafts and journals.

Energy lost by friction in ft.-lb. per min., = 0.2618 fwdn.

f = coefficient of friction;
w = weight on journal in lb.;
d = diaemter in inches;

TABLE III.—VALUES OF COEFFICIENT OF FRICTION, F.

Rubbing Speed Feet p.m.	209	262	314	360	419	471	Nominal load lb. per sq. in.
Coeff. of Friction....	.0010	.0012	.0013	.0014	.0015	.0017	520
Friction....	.0013	.0014	.0015	.0017	.0018	.002	468
Friction....	.0014	.0015	.0017	.0019	.0021	.0024	415

n = number of r.p.m.
Rubbing speed = 0.2618 dn.
Load per sq. in. = w ÷ ld; l = length of bearing, in.

Shaft Friction

Tables IV and V are based on experiments and tests made by Messrs. McAllister and Morley of the Case School of Applied Science. (Transaction A. S. M. E., Vol. 18). Indicator cards were taken from the engines during the day, at intervals of about 1 hr., while the plants were in full operation, during the noon hour and after working hours at night. Cards were taken from the same engine when it was driving line and countershafts alone.

Averages of these sets of cards were to show respectively the total horsepower and the friction horsepower. Length of bearings and diameter of shafts were measured, the number of bearings taken and the methods of oiling. The belts running from line shafts were counted, their width estimated and average diameter of pulleys taken.

Data obtained are given in the tables.

Lubrication

One explanation of the large loss by friction in many cases is the fact that economizing in either the quantity or quality of oil has at once a favorable effect on the bills, while the corresponding increase in coal and water consumption may be unnoticed or attributed to other causes. It is the opinion of Prof. J. J. Flather, based on tests, that a reasonable per cent of loss in shafting, for average light work with shafting in fair condition, is from 30 to 40 per cent of the total power expended, while for heavy work this loss will probably average from 40 to 50 per cent, assuming of course that the shafting is in fair alinement and properly lubricated, with its hangers sufficiently close to each other to prevent undue deflection under working conditions; also that the belting is not unnecessarily tight. Departure from these conditions will increase or decrease the assumed friction loss, depending upon the degree of perfection attained in erection and maintenance of the shafting. Knowing the results of tests made and the opinions of expert mechanical engineers based upon tests, it is the business of the engineer in charge to see that the shafting does not waste power, which is a waste of fuel and consequently of money.

TABLE V. RESULTS OF TESTS ON FRICTION OF SHAFTING

No. of Test	Total HP. Required	HP. Required for Shafting Alone	Per cent of HP. Consumed by Shafting
1	400	157	39.2
2	74	57	77
3	38.6	25.3	65.6
4	59.2	47.9	80.7
5	112	64	57
6	168	91	54.2
7	40.4	20.7	51.2
8	74.3	40	53.8
9	47.2	24.5	51.8

FIG. 1. LEVELING ROD FOR SHAFTING

FIG. 2. KINKEAD SHAFT LINING DEVICE

TABLE IV. DATA ON TESTS OF SHAFT FRICTION

No. of Test	Nature of Work done in Shop or Factory	Total length of shaft in ft.	Diam. of line shaft in in.	Speed in R.P.M.	Number of Bearings	Number of Belts	Average width of Belts in in.	Number of Counters
1	Wire Drawing and Polishing.....	1130	2½-2¾ 4 -6	170	115	89	4	69
2	Steel Stamping and Polishing.....	580	3 -3½	200	68	28	6	27
3	Boiler and Machine Works.....	530	2½-3	150	46	53	5½	47
4	Bridge Machinery.....	1460	2½-3 4	110	142	92	4½	79
5	Heavy Machine Work.....	1120	3	190	110	141	4	96
6	Heavy Machine Work.....	1065	2 -3 4	180. 150 180	114	192	4	152
7	Light Machine Work.....	748	1½-1¾ 2 -3	135-135 135-150	101	217	3	133
8	Manufacturing Small Tools.....	500	2 -3	114	58	235	3	214
9	Manufacturing Small Tools.....	990	2½-1¾	175-136	102	217	3	202

Alinement

Poor alinement may be due to changing the location of heavy machinery, uneven settling of floors, uneven wear of bearings, shafting supports becoming loose, bad adjustment of hangers, excessive spans between hangers or excessive belt tension.

Methods adopted for establishing a line of shaft or connecting the alinement must be worked out to suit the individual case. A method commonly employed is to stretch a wire to which the line of shafting can be compared. This reference line is usually placed directly over the shaft, its level being accurately determined by means of a spirit level, and its position in regard to the machinery and building carefully adjusted.

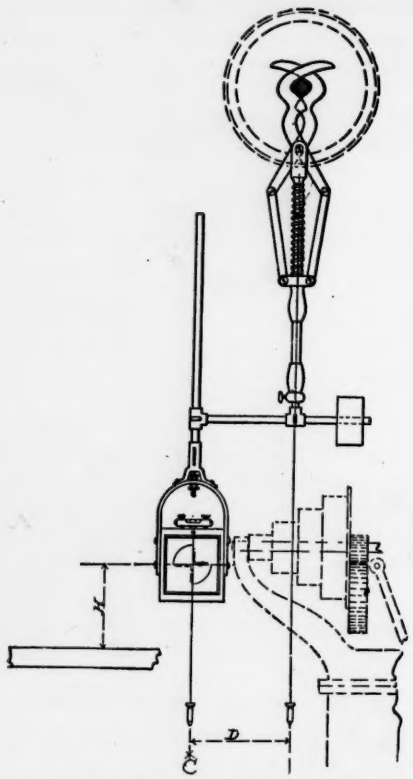


FIG. 3. OFFSET TARGET AND BALANCING DEVICE

Having the line accurately placed, measurements are taken both horizontally and vertically from the line to the shafting, and by means of the screw adjustment in the shafting supports, accurate alinement can be secured. Vertical alinement can be checked by means of a leveling rod and carpenter's level as shown in Fig. 1 and the horizontal line can be checked by hanging plumb bobs from various points along the shaft.

In taking the level of a shaft, a straightedge should be provided which will extend from one bearing to the next and the level applied to the straightedge, not to the shaft.

Another method of testing alinement of shafting is what is known as the Kinkead system. It consists essentially of a special architect's level built for indoor work and capable of sighting clearly on a line shaft 600 ft. long; a self-centering target which is hung fr the shafting by a jaw clamp so constructed that it n only finds the center of the shafting, but remains ce tered regardless of the diameter of shafting, and a fix or stationary target used as a check to insure unifor readings and results. The portable target can be a justed to enable the operator to work on shaftin whether overhead, in pits, on piers, under tables, wi or without obstructions under or over it. The faces the targets are transparent, and by the use of spec lanterns furnished with the outfit, the work can be do at night as well as in daytime. The outfit also contai an offset target, enabling one to work without removi belts.

Figure 2 shows the portable target as it hangs fr an overhead shaft, and the ordinary method of using t outfit. Figure 3 shows the offset target in use with b ancing device for bringing target in plumb.

A line shaft may appear to be properly lined when standing still, but upon revolving prove to be o

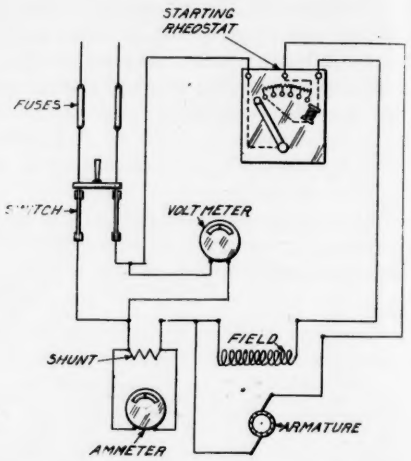


FIG. 4. CONNECTIONS FOR TESTING SHAFT FRICTION BY MOTOR

of alinement due to loose hangers; vibration, due to u balanced couplings, excessive belt tension which increas as load is applied or end play in shafting. A line sh can be tested while running by establishing a wire already described and measuring the distance at vario points to ascertain true running, also by a steady r such as a board fixed near the shaft and a piece chalk held to the shaft with the hand resting in a fix position on the steady rest. Character of the cha mark will indicate high spots, if any. Should the sha ing prove to be out of true, the cause should be soug out and corrected.

In many cases, excessive belt tension can be trac to improper care of belts, meaning that belts are lowed to become dry and slip; naturally the belts w be tightened to eliminate the slipping. In case of b bearings they should be immediately replaced by go ones or rebabbited. If hangers are loose, they shou be put in place and tightened. If belts are dry and t tight, they should be treated as explained in anoth article and have just enough tension to pull the lo

without slipping. If shafting is found to vibrate excessively, due to unbalanced couplings, those couplings should be taken down, put in a lathe and trued up before replacing them. In case of excessive end play, collars should be placed at proper positions to prevent this. Should the shafting be found to be bent, it should be taken down at once and straightened.

Arrangement of Pulleys, Clutches and Couplings

To secure the best service from belts and pulleys, and take advantage of the slack in the belt to give a greater arc of contact on the pulleys, they should be placed with their shafts parallel and with the tight side of the belt at the bottom. Vertical belts should never be used unless absolutely necessary, as any slack in the belt lets it hang away from the lower pulley, causing slip. This necessitates high tension, increasing friction, with an overpressure on the upper shaft bearings and possible lifting of the lower shaft so as to wear on the upper part of the bearings. If a vertical drive is unavoidable, a tightener pulley may be used to get traction with reduced tension.

A driven pulley located upon a long line shaft which has several driving pulleys should be located as near to the center of the load as possible. In this way the diameter of the shaft can be reduced considerably, because half the power of the entire shaft is transmitted in each direction, and the vibration also, which is likely to occur in long shafts, is reduced to a minimum. But a heavy belt should always have its pulley placed near a bearing to avoid springing the shaft.

Cutting Out Idle Lines

In the distribution of power by line shafting, economy can be obtained by providing for flexibility by the use of friction clutches or clutch couplings to cut off line shafts running idle or not in use at all times. Each department, line or shaft or individual machine may be controlled independently of the rest of the equipment. There results a great saving in power and expense by being able to cut out at will sections of a shop or nonproducing machinery. The proper places to install clutch couplings or friction clutches are where machines are idle a good part of the time, sections of shafting that carry little and infrequent load.

Testing for Friction Losses

Tests can be made for friction loss, to determine the percentage of power absorbed by friction of line and counter shafting, as follows: Take indicator cards on the engine at intervals during the day while machines are in full operation, and again at noon and at night when machines are not in use and the belts on loose pulleys. By averaging the indicated horsepower when running under full load and again when running light, and deducting the latter from the former, a very accurate record of the amount of power absorver by friction can be obtained.

For instance, if the power required to run machines at full load is 50 hp. and the power required to run idle is 10 hp. the friction loss will be as 10 is to 50, or 20 per cent of the total power required.

In the case of motor drives, the input in electrical energy when loaded and when idle must be compared. The proper readings to be taken and instruments used to take readings are, in the case of direct current motors, a voltmeter and an ammeter. Connect volt meter across the line supplying current to the motor and the ammeter in series with the motor as shown in Fig. 4. Care must be taken that the ammeter is of sufficient capacity to measure the full load current of the motor. The voltmeter reading multiplied by the ammeter reading divided by 746 will give the horsepower input of the motor. For instance, assume the readings to be 220 v., 170 amp. when running at full load; and when running idle, 220 v. and 32 amp.; $220 \times 170 = 37{,}400$ w.; $\div 746 =$ approximately 50 hp. when fully loaded; $220 \times 32 = 7040$ w.; $\div 746 =$ approximately 9.4 hp. The 7040 friction loss will be as 9.4 is to 50 or 18.8 per cent of the total power required. This percentage is low for shafting, but can be realized. Tests have shown that from 40 to 75 per cent of power is often wasted in shafting, belts, pulleys and clutches.

For alternating current motors an indicating wattmeter should be used and the readings taken and compared and explained for direct current motors. Having determined the friction loss, the next step is to find the cause, in case the loss is too high.

If it is found that the power loss in shafting is too great, locate where the most power is consumed. This can be done by making tests of various sections of the shafting. It may be necessary to uncouple the shaft in order to make a sectional test, but this must be done, and when it is found to be taking too much power the cause must be run down.

At a cost of nearly a million dollars the Potomac Electric Power Co. has installed a 15,000-kw. generator at its Benning plant, the turbine, with a 21,000-hp. capacity, having recently been placed in service.

The new unit alone has sufficient power to operate all the street cars in the district. When combined with the former equipment, it gives Washington a power plant said to be second to none in the country.

An order for the machine was placed with Westinghouse Co., Dec. 11, 1916. It was delivered Sept. 1 last. It gives the Benning station a total capacity of 67,000-kw., or more than 110,000 hp.

The company, in October, 1917, ordered four boilers, and in June, 1918, ordered two more of 1000 hp. each. Two of the boilers have been installed and are in operation.

In order to accommodate the new turbine it was necessary to erect a large annex to the Benning plant. This is equipped with two stacks, each 205 ft. in height and 12 ft. in diameter.

From statistics of the U. S. Employment Service it appears that requests from employers for workers have been declining as compared with registrations for positions in most states in the country since about the middle of October. Before that the positions exceeded applicants in ratios of 2 to 1 up to 18 to 1 for various states. At present the average call for men about equals or is somewhat less than the supply.

Inspection and Test of Electric Transmission Lines

Troubles and Losses in Overhead Transmission. Tests and Remedies to Get Best Efficiency. By O. C. Callow

ONE OF the first necessities for efficient operation of a transmission line is a complete inspection of all apparatus, to be made at specific times, a complete written report being kept of all faulty material and the repairs made. When the climatic conditions under which a transmission line has to operate are considered, varying from summer heat to winter snow and ice, it can readily be seen that inspection is one of the chief things necessary to insure against shut-down.

Pole Lines

All poles should be inspected, if of wood, to locate any rotting at the butt or bad cracks, those poles which are in bad condition being replaced. If the poles are of the built-up steel type, they should be painted at any places showing a tendency to rust. All guys should be inspected for breaks or rusting and should be tightened when required. All insulators should be gone over carefully to locate any which have become cracked or loose, and a test should be made of each for leakage to ground. On a high-voltage line, faulty insulators can often be located in damp weather, by the static discharge, which can be heard in the daytime or seen at night. Faulty insulators will soon cause serious trouble on the line, and will often burn the cross arms of wooden poles, particularly if two insulators become grounded on one arm. On steel poles a short circuit will occur, putting the line out of action and probably burning the cable, making a large repair job.

All insulators should be kept clean; this is particularly important if the line runs through certain industrial plants, such as cement or lime works, where a layer of cement or material of like nature will form over the insulators causing a large surface leakage in wet weather, which may in time cause the insulator to break down. This deposit should be washed off with dilute hydrochloric acid or similar solution.

Cables should be inspected where they are attached to the insulators, all clamps being tightened up and any signs of wear noted. Sag between spans should be equal on all cables to prevent danger of their touching in a high wind. There is no definite rule to determine the correct sag of the cable as there are so many factors which must be taken into consideration, such as the size and type of cable, length of span, and climate.

All disconnect switches located on poles or outside substations should be inspected, all contacts tightened up and cleaned and coated with vaseline. These switches should be located in such a position that they can be easily opened by the operator when the line is alive without danger to him, even if some load is on the line. Oil switches should be installed at all points where it will be necessary to switch off load on the line.

The line should be well protected against lightning by some approved form of lightning arrester. Too much attention cannot be paid to this point as a large amount of damage may be done to the apparatus in the power plant through the failure of arresters to clear the li during an electrical storm. The repair of this apparat would probably cost much more than a complete i stallation of high grade arresters. If the arresters a of the electrolytic type they should be charged eve day and the color of the charging spark noted; if t films are in good condition, this spark should be of bluish tint; if it has a red color, the arresters shou be charged longer. The horn gaps should be carefull adjusted and should be equal on all phases. A sm amount of the electrolyte should be drawn from t tanks each month and tested. The ground wires fr the arresters should be inspected and all connecti tightened up. All ground wires on poles should al be gone over.

Line Tests

Voltage drop is an important factor which shou be checked up, as it will show whether losses are exc sive. This can be tested by connecting a voltmeter one end of the line, and a voltmeter and wattmeter the generation station. Simultaneous readings shou be taken on all three meters, the difference between t voltmeter readings giving the drop at the various loa noted on the wattmeter. This figure should be about per cent at full load on the line. If it is found that t drop is high, it is proof that there is excessive line lo which may be caused by grounds or overloading.

It is useful to obtain a record of the losses on t line during certain fixed periods of time, to show an large increase in the losses and act as a guide to troubl This test can be made by installing a watt-hour met at each end of the line, provided there are no branc circuits; the difference in readings taken every 12 h will be the total loss on the line during that perio From this can be calculated the percentage loss per hou giving a definite check on the condition of the line if it is found that the average losses increase, it is good indication that there is some trouble on the lin which is causing leakage and an investigation shoul be made at once.

Low power factor is the indirect cause of muc trouble on transmission lines since the current in th conductors must be increased, if the output is to be main tained the same with a low power factor as it would b for the higher power factor for which the line was de signed. Increased current in wires may mean overheat ing, higher resistance and greater voltage drop, whic will again tend to increase the current. Under thes conditions the I^2R losses will be high and the efficienc of the line poor.

If it is found that the losses on the line are steadil increasing, a test should be made to check up the powe factor. If a power-factor meter or reactance meter i installed on the power house switchboard, this is an eas matter and a series of readings should be taken showin power factor at various loads. If a meter of this kin

is not available, the test can be made by connecting a standard voltmeter, ammeter and wattmeter in the circuit and taking simultaneous readings on all three meters. The power factor can then be found by dividing the wattmeter reading by the product of the ammeter and voltmeter readings. This should be repeated at various loads and should give a power factor of from 70 to 80 per cent, depending upon the nature of the connected load. As most transmission lines operate at a voltage too high to be used direct on meters, it is generally necessary to use transformers for all three meters, which somewhat complicates the connections, and care should be taken to get the correct relation between current and voltage coils on the wattmeter. The correct connections for this test are shown in the illustration.

As stated above, the power factor of the system is fixed by the nature of the connected load which it supplies, therefore any remedy to improve this power factor must be applied at the distribution end of the line and not in the power house. If the load consists chiefly of induction motors, the power factor should be about 80 per cent when the motors are fully loaded. It will fall much below this figure, if the motors are only partly loaded or running idle on the line, owing to the fact that the magnetizing current of an induction motor is out of phase with the current utilized for power and is nearly constant at all loads. Practically the same condition exists in the case of transformers, with the addition that their regulation is seriously affected by low power factor.

Raising the Power Factor

If it is found that the power factor is bad, a careful study should be made of the apparatus connected to the line, with the object of distributing the load so that motors and transformers will be fully loaded, or shut down when not in use. A considerable improvement can be made, if some of the motor load be replaced by a lighting load, which being noninductive, will increase the resultant power factor.

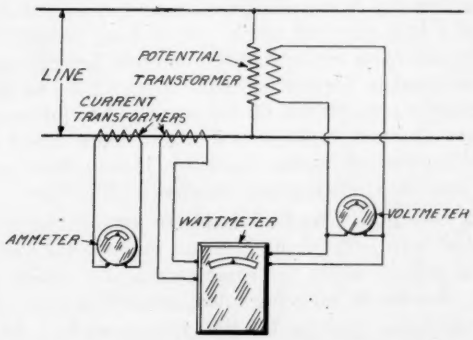
In many cases, however, it is found impractical to improve the power factor by rearrangement of load as it involves too much change in existing plant. In these cases, a good solution is to install a synchronous condenser at some point near the center of heaviest load. This will consist of a synchronous motor arranged so that its field can be over excited. This machine will deliver a leading current to the system, to compensate for the inductive load produced by the induction motors; in other words, it will supply the magnetizing current to the load while the power component is supplied by the generating station, so that the line carries no wattless current. The synchronous motor can be used to drive any mechanical load requiring to run over long periods with a steady output, such as an air compressor or pump, and will still fulfill its functions as a condenser. The best results will be obtained with the motor carrying a mechanical load of about ⅔ of its rated output.

When the saving in copper on the transmission lines together with the lower losses and better regulation is taken into consideration, it will be found that the installation of a synchronous condenser is more than justified on any line where the power factor is permanently low. The following example is of interest to show the great difference in size of conductors required on a line with high and low power factor.

A line is required to transmit 1000 kw. at 6000 v. 3-phase over a distance of 5 mi. with a drop not over 10 per cent. If a power factor of 90 per cent can be maintained, each conductor would have an area about 97,500 circular mil., but if the power factor were only 60 per cent, each conductor would have to be about 218,000 circular mil. or 2.2 times as heavy as with the higher power factor.

Line Insulation

Arrangement should be made in the generating station so that the general condition of the line insulation be known and kept on record. This can be accomplished by means of a ground detector. On a high-voltage system which is not permanently grounded, this instrument is generally of the electrostatic type, the reading being based upon the measurement of potential difference between each line and ground. The line showing the smallest reading will therefore be the one which has the poorest insulation.



CORRECT CONNECTION FOR POWER-FACTOR TEST BY VOLTMETER, AMMETER AND WATTMETER ON HIGH-VOLTAGE CIRCUITS

On systems having their neutral point grounded at the station, an ammeter can be connected in series with the ground wire. This meter will read zero when there is no current flowing, thus indicating that there is no leakage to ground. Should a ground occur, the meter will show a reading and the higher this reading, the worse the ground. These readings should not be used to base calculations upon, as they will be affected by many outside influences such as capacity, etc.; they are, however, useful for showing the general condition of the insulation of the lines. If a bad ground be noted through the reading on the detector, it can sometimes be located by opening the disconnect switches on different sections of the line until it is noted that the meter again reads zero; the last section disconnected will be the one which is grounded, and a careful inspection of this section should reveal the trouble.

A megger can be used for testing out the insulators for leakage to ground; but is not altogether reliable, inasmuch as some insulators will show less resistance due to dirt than others which show a high test, but are really

defective and will fail on a dielectric test. Generally, if the weather is dry, the insulators which show a low resistance will be found defective upon a close inspection and will fail on a dielectric test. A telephone receiver can be used to locate faulty insulators by connecting it in series with the insulator pin and the ground wire on the pole. A faulty insulator will give a different indication from a sound one. This scheme can only be used on poles having wooden arms as on steel poles the insulator pin is already grounded.

Rearranging Load

In many cases a considerable economy can be secured by making a study of the lines with a view of rearrangement to meet present conditions of load, which may have changed from those when originally laid out, due to unexpected rapid growth of consumers in new districts, causing overload lines and large voltage drop. Inasmuch as the voltage drop on the line is practically proportional to the distance over which power is transmitted, an endeavor should be made to keep the distance between the generating station and the center of load as small as possible by running the lines as direct as local conditions will allow.

It can readily be seen that the losses on an overloaded and badly laid out line taken over a long period, will cost a considerable amount of money, which could much better be spent in improving conditions on the line, such as running a new circuit on the same poles, putting in heavier conductors, building a new and more direct line or installing a synchronous condenser, if the power factor is low, and thus cutting out the losses altogether.

It is sometimes found advisable to install new transformers at both ends of the line and increase the voltage over the system, using the same conductors, which will then be capable of carrying much greater power. It is always advisable to keep the line voltage as high as the distance of transmission will allow, thus making a saving in copper in the conductors. Modern transformers and insulators are quite capable of taking care of practically any voltage up to 100,000 and the cost of the higher voltage line equipment is more than offset by the copper saving, if the distance is long. Another disadvantage of the low-voltage line is, that as the size of the conductors is increased, there will be a certain amount of skin effect which will cause increased drop; also the I^2R losses will be greater when a large current has to be transmitted.

Line Transposition

In designing or rearranging a transmission line, particular attention should be paid to the transposition of the conductors of the line in order to equalize the electromagnetic induction of the conductors with any neighboring circuits. In order to accomplish this, the distance from any outside circuit must be the same to each conductor of the line. The conductors should therefore be transposed in such a manner as to make the number of transpositions on the total length of the line a multiple of three. This is particularly necessary on a high-voltage line which runs in parallel with other lines and which may cause quite an appreciable loss. If two or more generating stations on the system are located some distance apart, having the main load adjacent to one of them, an improvement in operation can be made by over-exciting the generators in the station which is situated nearest the load. This will cause the machines in this station to supply a large amount of the lagging current which is taken by the load. In this way, the losses due to reactive current flowing along the line from the far station will be minimized, thus increasing the efficiency.

Transformer Arrangement

A careful study should be made of the distribution of load on all transformers on the system with a view of keeping them fully loaded and so cut down their losses. Switches should be provided to disconnect any transformers from the lines which are not actually in use, such as those installed for plants which operate only at certain seasons of the year and those which would only be required in the winter months. The iron and magnetizing losses on these transformers will be considerable, even at no load and will cause low power factor on account of induction.

Great saving can be made by the use of large capacity transformers in place of a number of smaller ones, as the total losses on the smallest units will amount to much more than the losses on a few fully loaded transformers. Furthermore, the performance of the large units is better in regard to power factor, efficiency and regulation. In some places where three transformers are in service, connected in delta, and carrying only a small load, it is advisable to reconnect two of the transformers in open delta, cutting the third out altogether or keeping it on hand as a spare, so arranged that it can be cut in with switches and take the place of one of the others in case of trouble or in addition to them, in case of heavier loads. This will do away with the losses of one transformer altogether and decrease those on the other two as they will be better loaded, resulting in increased overall efficiency and power factor and better regulation.

Disconnections

All lines should be provided with disconnecting switches so arranged that any sections of the line which are not in use can be cut out from the system and thus do away with all losses on them. A high-voltage line has considerable condenser effect, which even at no load, will amount to quite a little wasted power. In addition, there may be losses due to faulty insulators to grounds or partial shorts, the saving of which will soon pay for the installation of the necessary switches.

In conclusion, a few words showing the importance of operating all transmission lines at their greatest efficiency will be in order. The generators in the power house are capable of developing only a certain limited output, which has to be distributed to the consumer by means of the transmission lines. If the efficiency of the lines is 100 per cent, there will be 100 per cent of the power for the consumer to use and to produce revenue for the generating company. If the efficiency of the line is only 50 per cent, the consumer has only half the power to use and the company receives only half the revenue, but still has to burn the same amount of coal and has the same operating expenses.

Transformer Losses

TESTS FOR CONDITION, INTERNAL LOSSES, AND DISTRIBUTION LOSSES. BY V. E. JOHNSON

LOSSES IN connection with transformers may be classified variously as: Internal and external, normal and abnormal, operating and maintenance, direct and consequential. It is not always possible to place a given loss under any one of these headings, for it may, and generally does, fall under several of them. Thus an abnormal loss may be internal, and may bring with it consequential losses.

An internal loss is one due to conditions existing in the active material of a transformer; for example, the copper loss.

An external loss is due to conditions existing in transformer connections, such as circulating currents due to improper taps being used in a bank.

A normal loss is one inherent in the design: Instances are copper and core loss.

An abnormal loss is due to unnatural conditions and improper operation. An example is loss caused by a short-circuit between turns.

Oprating losses are those which result from operation and are so named to distinguish them from—

Maintenance losses, which are those incurred when the transformer itself is damaged or destroyed.

Direct losses are those occurring in the installation itself while;

Consequential losses are those which arise outside of the transformers themselves—such as damage to persons or machinery.

The above tabulation is not in any sense an outline of this article, but is intended as a survey of the field, with the hope that it may serve to impress upon the operating man the importance of knowing exactly the condition of his transformer system. Even losses which in themselves are insignificant, loom large when one considers that from them may come unsatisfactory operation, with its attendant reduction of income.

CONNECTIONS

STANDARDIZATION of electrical apparatus agreed upon by the manufacturers has to a large degree eliminated the simpler errors of connection. Years ago, improper connections due to mistaken polarity were not uncommon, and in the majority of cases did no damage beyond the blowing of a fuse or the interruption of service. Since the adoption of a standard method of bringing out leads from the coils through the case such mistakes are rare.

Figure 1 shows the usual relations between leads in an ordinary distributing transformer.

Parallel operation of transformers requires

(a) Proper polarity.
(b) The same ratio of transformation.
(c) Similar regulation characteristics.

No further consideration need be given the matter of polarity. Errors in ratio must, however, be guarded against rather carefully as mistakes here are not uncommon. It is obvious that even a small difference in the voltage of coils, operated in multiple, with short, low-resistance connecting leads would result in heavy circulating currents and the accompanying losses and heating.

In rare instances, the manufacturer supplies an incorrect name plate, but misreading of the data stamped on it causes the great majority of such errors. Companies which use both standard 2200 to 220-110-volt transformers and 2200 to 244-122-volt (now nonstandard) transformers are very likely to encounter trouble from "mixing" these two ratios. This mistake is particularly likely to happen in the case of a syndicate shipping apparatus to its various properties from a central warehouse.

In banking transformers provided with taps, care

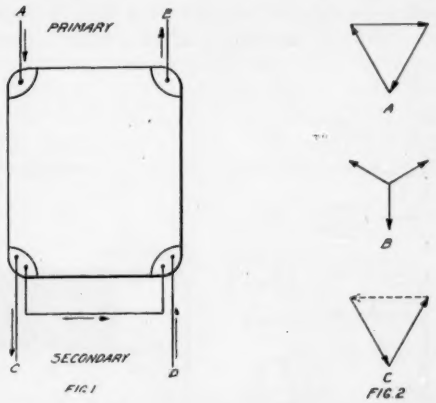


FIG. 1. USUAL ARRANGEMENT OF TRANSFORMER LEADS. ARROWS INDICATE RELATIVE INSTANTANEOUS POLARITIES. TOP VIEW WITH COVER OFF

FIG. 2. CONNECTION DIAGRAMS FOR TRANSFORMERS: A, FOR DELTA OPERATION; B, FOR STAR CONNECTION; C, FOR OPEN DELTA

should be taken to connect similar taps, as the percentages of error involved may cause serious heating. It is advisable always to refer to connection diagrams in such cases, for even the most expert lineman cannot distinguish a 2½ per cent tap from a 5 per cent tap by appearance or location alone.

Due to differences in design, the regulation curves of the various makes of apparatus are often unlike. The result is that when attempts to parallel them are made, the load is not divided properly. A few of the leading manufacturers have practically identical design, and no transformers in any combination; but as a general rule it is well to consult the makers before connecting important installations consisting of more than one type of apparatus. Methods of experimentally deriving the regulation curve of a transformer will be described later.

In polyphase systems, the number of possible combinations is so large that it is impossible to list them all. There are, however, a variety of simple schemes for

keeping these combinations in mind, of which the following is perhaps the simplest:

1st. Connect each transformer series, series parallel or parallel, leaving out of consideration entirely the interconnections between transformers (which are to be made later).

2nd. Reduce (on paper) each transformer to its simplest terms, i. e., neglecting all internal connections consider it as simply a two-coil transformer with two primary and two secondary leads.

3rd. Indicate or represent each of these elementary coils by an arrow, the tail representing the start of the coil, the point the finish. In other words, the direction of the arrow indicates the polarity of the coils.

4th. For delta operation use the plan of Fig. 2A. The three arrows follow each other, around the triangle, expressing simply the rule that in this connection the beginning of each coil is joined to the finish of the next, these junctions being the terminal points for the main leads.

5th. For star or Y operation use connection 2B. This expresses simply the rule that in a Y connection, the corresponding terminals of the three transformers

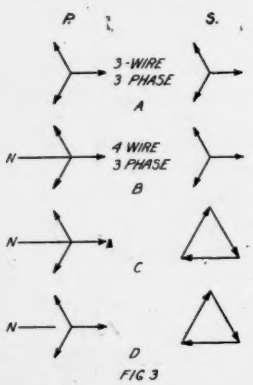


FIG. 3. COMBINATIONS OF WINDINGS FOR PRIMARY AND SECONDARY: A, 3-WIRE, 3-PHASE; B, 3-PHASE, 4-WIRE PRIMARY, 3-PHASE SECONDARY; C, 4-WIRE PRIMARY STAR, SECONDARY DELTA; D, 3-WIRE PRIMARY STAR, SECONDARY DELTA

FIG. 4. WET PAPER FOR DETECTING WATER IN OIL

are joined, the line leads being attached to the three free terminals.

6th. Open delta is the result of removing one transformer from the delta and leaving the polarity relations of the other two unchanged, as in 2C. The dotted arrow shows the missing transformer.

With a few exceptions of which the most notable are indicated in Fig. 3, any combination of star or delta primary to star or delta secondary is workable.

A may cause unbalanced voltages in the secondary, analogous to a blown neutral fuse in a 3-wire system.

By using a 4-wire primary, B (connecting the neutral point to neutral of generator), this unbalancing is avoided.

C will not work on a 4-wire system because of the setting up of harmonics in the secondary, resulting in heating which may damage the insulation.

By cutting off the neutral wire as at D, this trouble is eliminated as it isolates the generator harmonics from the transformers.

(In the above statements reference has been made to harmonics. It is not possible to discuss these phenomena in an article of this size; the reader is referred to any standard advanced text book.)

Errors in the connections of transformers on polyphase systems usually manifest themselves by unbalanced or improper voltages at the receiving end, and losses resulting from these are usually consequential. Motors may fail to operate properly or may burn out, and various similar connected apparatus may give dissatisfaction to the user.

Oil and Cooling Systems

Oil in a transformer serves two purposes, first, it carries the heat generated in the interior of the transformer to the outside, where it is transferred to the case or to the cooling coils, and second, it serves as an insulating medium.

In carrying out its first function, that of cooling, the oil circulates through the various ducts between the coils. It is obvious that dirty oil will not pass freely through these passages, and will deposit in them foreign matter

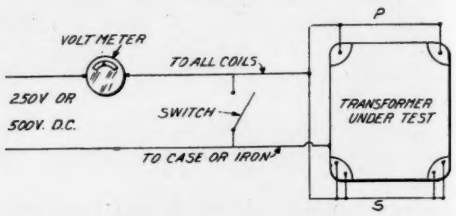


FIG. 5. CONNECTIONS FOR INSULATION RESISTANCE TEST OF TRANSFORMER

which hinders heat transference, both directly and by still further retarding circulation.

In order to act as an insulator, it is necessary that there be no water in the oil. In 2200-volt transformers (and in those of still lower voltage) small quantities may be present without doing any harm, but in the higher potentials, even minute amounts will, under certain conditions, cause burnouts.

Dirt in oil is readily detected; water can be discovered by the following methods: (1) Wetting a piece of soft paper partly in the oil, the water will reveal itself in a narrow line between the oiled and dry portions of the paper, Fig. 4.

(2) Plunging an iron rod heated to a faint red into the oil indicates the presence of water by a crackling sound.

(3) Using an oil-testing outfit consisting of spark gap and high-voltage testing transformer. The standard break-down test is 40,000 volts across a 0.15-in. gap between ½-in. spheres, with the gap 6 or 7 in. below the surface of the oil and the temperature about 70 deg. This is a very accurate test and also locates other impurities such as carbon particles and suspended metal dust. Percentages of water far too small to be indicated by tests 1 and 2 are readily located by this method.

If the number of transformers in service justifies the expense, the last method is ideal; but it involves an expenditure for a complete outfit of some $500.

Water having been detected, there is, strictly speaking, only one method of removing it, and that is by means of an oil filter. Heating the oil in suitable metal containers will, it is true, drive off the moisture, but there is grave danger of overheating and damaging the oil itself. Furthermore, any heating process removes only the water; particles of metal and carbon remain in suspension and are not affected.

Water finds its way into a transformer case in several ways. If cold oil is handled in a warm room, moisture will condense into it. If a transformer case be not airtight, moist air may enter it and eventually cause water to accumulate. A cracked cover will allow rain to enter.

In water-cooled transformers leaks sometimes develop in the cooling coils. But a more usual source of moisture on these types is the sweating of the coils above the surface of the oil. It is customary to lag or insulate the upper turns of pipe to prevent this, and care should be taken that all of the exposed pipe is so covered.

New transformers when of the larger sizes may gather moisture in shipment, as it is often necessary to ship these outside of the case. There is no certain method of knowing whether an appreciable amount of water is present, but the nearest approximation available is obtained by making an insulation resistance test. This is done by taking readings, using 250 or 500 volts direct-current, with a high-resistance voltmeter. Figure 5 shows how these readings are taken. Closing the switch gives the value V, opening the switch gives v. Primary and secondary leads are connected together during the test.

By using the following formula the insulation resistance is obtained:

$$R = \frac{r\,(V - v)}{v}$$

Where, R = insulation resistance in ohms; r = resistance of voltmeter used (as marked in the cover or on the scale); V = voltage of line; v = voltage indicated on voltmeter with insulation in series. R should not be less than 300 megohms. If it tests less than that value, the coils must be dried out before being placed in the tank.

This can be done in several ways, the two simplest being: (a) Short-circuiting low tension leads and applying a low voltage to the high potential coils until the temperature of the transformer is maintained at from 75 to 80 deg. centigrade (165 to 175 deg. F.), as determined by thermometers placed on various parts of the apparatus. (b) By passing heated air from a hot air system over the coils, maintaining temperatures as in method (a).

Insulation resistance should be tested hourly and the results plotted as a curve until the readings become nearly constant. Then, as an added precaution, the drying process should be continued 1/5 of the number of hours required to reach this point. That is, if 20 hr. were required to reach constancy, 4 hr. more should be allowed as a margin of safety.

On large transformers (even though they are in perfect condition when installed) a sample of oil should be drawn off from the bottom of the tank every four or five months and tested.

Water-cooled transformers have little radiating surface outside of the cooling coils; any accumulation of scale inside of these or of sludge or slime on the outside results in an immediate rise of temperature. Good transformer oil will not deposit any sludge, but poor circulating water will produce various kinds of scale which must be removed by acid solutions. When a water-cooled transformer is to be stored away or left idle indefinitely, the coils should be drained of water and filled with oil.

Internal Tests

So far we have dealt only with conditions existing outside of the coils and iron. Of the internal tests, the most important is that of the insulation. Hundreds of thousands of people are daily trusting their lives to the insulation between the primary and secondary coils of the transformers which supply them with energy. Grounding of secondaries has reduced this "personal" danger, but should not serve as an excuse for shirking the rigid inspection of major transformer insulation.

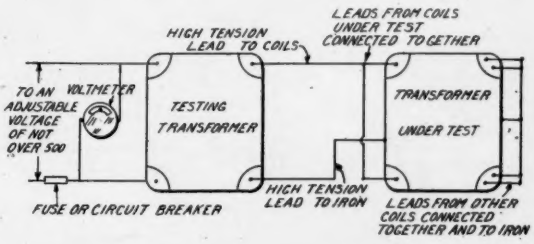

LEADS FROM COILS UNDER TEST CONNECTED TOGETHER
HIGH TENSION LEAD TO COILS
TO AN ADJUSTABLE VOLTAGE OF NOT OVER 500
VOLTMETER
TESTING TRANSFORMER
TRANSFORMER UNDER TEST
FUSE OR CIRCUIT BREAKER
HIGH TENSION LEAD TO IRON
LEADS FROM OTHER COILS CONNECTED TOGETHER AND TO IRON

FIG. 6. CONNECTIONS FOR TESTING COIL INSULATION OF A TRANSFORMER

The insulation resistance test as described in a previous paragraph is valueless in determining the dielectric strength of barriers in transformers. For making these tests, 1, source of high voltage; 2, means for adjusting this voltage, and 3, means for measuring it are necessary.

The source may be a high-tension transformer of voltage as indicated in the table of test values, on which the low voltage can be varied, either by means of an induction regulator, or by varying the excitation of the supplying generator. The voltage can be measured by a voltmeter connected to the low-tension wires of the transformer, or by means of a needle gap. The former method is simpler, and is perfectly satisfactory, except for extremely high potentials. Figure 6 shows how connections are made for the test.

The following table taken from the A. I. E. E. rules shows test voltages to be applied. The potential should be raised gradually to the required value, held there for one minute, and then reduced to a low value before opening the switch.

TABLE I. TEST VOLTAGES FOR INSULATION TEST

Voltage of Coils	Capacity	Test Voltage
Under 400 volts	Under 10 kw.	1000
400-800	Under 10 kw.	1500
Under 400	Over 10 kw.	1500
400-800	Over 10 kw.	2000
800-1200	Any kw.	3500
1200-2500	Any kw.	5000
Over 2500		double rated voltage

Transformers having 550 to 5000-volt primaries for distribution service should have 10,000 v. applied between primary and ground and between primary and secondary. Between secondary and ground use the voltages in the table.

CORE LOSS

CORE loss is that due to the continuous reversal of the magnetism in the iron of the transformer. It goes on as long as the transformer is connected to supply line, and is practically constant, regardless of load.

In Fig. 7 are shown the proper connections for "running" a core loss curve. The testing current is, for convenience and safety, applied to the low-tension windings, though the same results could be obtained by supplying power to the high potential side. The coils to which current is supplied can be connected in series,

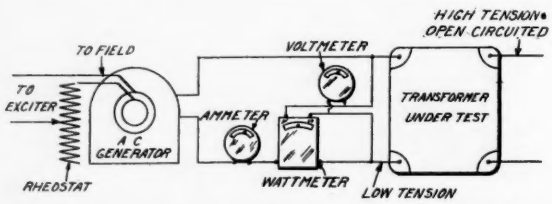


FIG. 7. CONNECTIONS FOR RUNNING CORE LOSS TEST

parallel, or series-parallel, without affecting results of the test.

The wattmeter should have a rating of approximately 2 per cent of the transformer capacity. Thus a 7.5-kw. transformer would require a 150-watt wattmeter. This proportion varies with different sizes and makes, but serves as an approximation. Similarly, the ammeter will be required to have a capacity in the neighborhood of 3 per cent of the ampere rating of the transformer, as connected for test. The voltmeter must, of course, be capable of indicating at least normal low-tension voltage and should have a scale such that ¾ normal voltage can be easily read. Figure 7 shows all self-contained instruments, but if the low-tension voltage is over 500 it will usually be necessary to use instrument transformers. This in no way alters the method of carrying out the tests, but for the sake of simplicity in this discussion a 220-110-volt secondary with its coils on multiple has been assumed as being under test.

After connections have been made as shown, a series of readings starting with ¾ voltage should be taken and the results tabulated as in the table, which is an actual curve taken in the testing department of a leading manufacturer.

The voltmeter had 1750 ohms resistance. Wattmeter had 3500 ohms resistance. The test was taken with secondary coils in parallel for 110-volt operation.

TABLE II. RESULTS OF A CORE LOSS TEST

			Instrument Loss		
Volts	Amps	Indicated Watts	Volt-meter	Watt-meter	Net Watts
85	1.0	52.0	4.1	2.0	46
95	1.10	57.0	5.2	2.6	49
100	1.20	61.0	5.7	2.8	52.5
105	1.30	65.0	6.3	3.1	56.0
110	1.43	70.0	6.8	3.4	60.00
115	1.58	75.0	7.6	3.8	64.00
120	1.76	80.0	8.2	4.1	68.00
125	2.03	88.0	9.0	4.5	75.00

Frequency normal at all points.

It is usually permissible to neglect the error due to power consumed in the instruments, but if tests are being made for the purpose of checking guarantees, they may be allowed for as follows: The wattmeter as connected includes in its indication the watts lost in the voltmeter and in the potential coil of the wattmeter. (In passing, it may be noted that in some types of wattmeters a special binding post is provided which automatically allows for the losses in the instrument itself, thus making it necessary to deduct only the voltmeter loss from its readings.) These values are determined by squaring the voltage applied and dividing by the

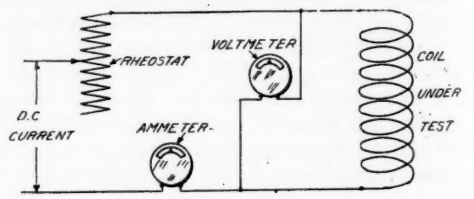


FIG. 8. CONNECTIONS FOR MEASURING RESISTANCE OF TRANSFORMER COIL

resistance (as indicated in its cover) of the instrument. Thus if the voltmeter had 2000 ohms resistance its loss at 120 volts would be $120^2 \div 2000 = 7.2$ watts. The sum of the voltmeter and wattmeter potential coil losses should be subtracted from the wattmeter readings. Ammeter readings have no bearing on the core loss; indicating as they do merely the magnetizing current, no allowance need be made for the extremely small currents taken by the potential coils of the volt and wattmeters.

Manufacturers' guarantees are based on a sine wave of impressed voltage and reference is always made to the temperature at which ratings hold, usually 25 deg. C. (77 deg. F.). It is difficult to determine the wave shape of the testing current without the use of expensive and complicated apparatus. As a result, core loss as referred to voltage measured by an ordinary voltmeter may vary by several per cent either way. To avoid this ever-present source of error, there has been placed on the market for use in transformer and apparatus testing a so-called iron loss voltmeter, which indicates, not the effective voltage, as does an ordinary voltmeter, but the value of a sine wave voltage producing the same loss as does the potential to which the instrument is connected.

If the iron-loss voltmeter, for example, were connected in Fig. 7 and indicated 100 volts, this would mean that the iron loss in the transformer under test was that of a 100-volt sine wave.

Copper Loss

This is the I^2R loss in the conductors of the primary and secondary coils. It varies with the load, becoming almost zero at no load.

To find this loss for any current, it is only necessary to multiply the measured secondary resistance by the square of the secondary current, and to add to this product the product of the primary resistance and the square of the primary current. Obviously then, the first step is to find the resistance of the transformer.

The drop of potential method is the most convenient one for obtaining resistance and is shown diagrammatically in Fig. 8.

Direct current is used for this test, and should be kept at a very low value (say 20 per cent of the normal rated current of the coil under test) in order that errors due to changes of temperature may not be introduced. Three readings at different points should be taken and tabulated as follows:

Table III. Resistance Test Values

Primary				Secondary			
			Average				Average
V	A	Ohms	Ohms	V	A	Ohms	Ohms
1.81	.424	4.25		.310	5.05	.0615	
2.62	.610	4.30	4.30	.369	6.00	.0615	.0615
3.12	.720	4.34		.430	7.00	.0615	

This test should be made quickly and the current interrupted between readings, in order to keep the temperature low. Resistance should not be taken until the transformer has reached the temperature of the room in which the test is made, in order that a thermometer placed on the coils may indicate not only temperature at the surface, but of the interior as well.

The resistances obtained by the foregoing method should be reduced to the resistances at standard 25 deg. centigrade by the following formula:

$$\text{Res. at 25 deg.} = R\left(\frac{263}{238+t}\right)$$

where R is the resistance as measured at temperature t in deg. C.

In making resistance tests, the size of the ammeter to be used is, of course, obtained from the name plate data. The voltmeter required can be calculated from the maker's efficiency data; but a double scale voltmeter, 0 to 1.5—0 to 15 volts, will be of correct size for any 2300-volt transformer.

Having determined both primary and secondary resistance at 25 deg. C., a tabulation as in Table V will serve as a data sheet on which to base a copper loss curve.

Short Circuit Test

This test is used to determine the regulation of the transformer without applying full loads to it, also as a check on the copper loss as determined by the resistance method.

The connections for this test are made as in Fig. 9.

If high accuracy is wanted, allowance must be made for instrument losses in the manner previously described. The size of the instruments can be determined by the "trial" method or can be calculated as follows:

Voltmeter should be designed to read between 2 and 5 per cent of the primary voltage. Ammeter should be rated at 150 per cent of the primary amperes of transformer under test. The wattmeter should be of approximately the size determined by the resistance test of copper loss, as it will indicate slightly more than the sum of primary and secondary copper losses.

Readings should be taken at ¼, ½, ¾ and full load current, and, if the ammeter is sufficiently large, at 1¼ and 1½ load. Tabulations should be made as in Table IV. Frequency must be kept normal.

Table IV. Values From Short Circuit Test

% Load	Impedance Volts	Amps.	Indicated Watts	Loss in Voltmeter	Loss in Wattmeter	Net Watts
25	13	.85	8.15	.10	.05	8
50	27	1.70	31.6	.41	.20	31
75	40	2.55	73.46	.97	.49	72
100	54	3.41	129.9	1.66	.83	127
125	66	4.26	198.7	2.50	1.25	195

Resistance of voltmeter 1750 ohms.
Resistance of wattmeter 3500 ohms.

On large transformers care should be taken that the leads used to short-circuit the secondary are heavy

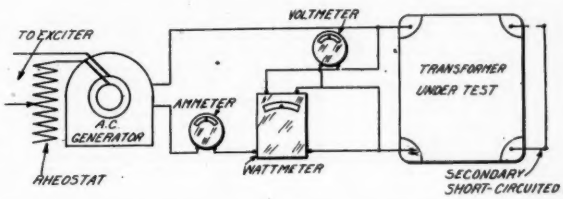


FIG. 9. SHORT-CIRCUIT TEST FOR DETERMINING REGULATION OF TRANSFORMERS

enough, as any losses in these connections will be included in the wattmeter readings and so impair the accuracy of the test.

If a comparison be made between the short-circuit curve wattmeter readings and the copper loss measured by the I^2R method, a slight difference will be noted. The former being larger. This is due to the fact that the wattmeter measures, in addition to primary and secondary copper loss, a slight core loss (which at 2 or 3 per cent of normal voltage is almost nothing) and the eddy currents in the conductors themselves. These eddy current losses may be segregated by subtracting from the wattmeter readings at any load the corresponding I^2R loss obtained by calculation. These values are sometimes called load losses.

Regulation

Regulation is the ratio of rise of secondary voltage from full load to no load, to full load secondary voltage, the primary voltage having been kept constant through-

out the tests. Experimentally, it can be determined by measuring full load voltage, V, and no load voltage, v (using a lamp or water rheostat load for 100 per cent power factor determination), and applying formula; regulation = (v — V) ÷ v reduced to per cent. This method is, however, inconvenient, and for large transformers entails heavy losses of power during the test, as well as the construction of cumbersome load devices. By using the data obtained in the impedance or short-circuit curve in the following calculations, regulation can be accurately and easily determined.

(A) Let IR = resistance drop of transformer expressed in per cent of rated voltage. This is obtained as follows:

$$\frac{\dfrac{\text{Primary IR}}{\text{ratio of transformation}} + \text{secondary IR}}{\text{rated secondary voltage.}}$$

Where I = rated full load current, and R = resistance as obtained in a previous test.

(B) Let IX = reactive drop expressed in per cent of rated voltage. It is found as follows:

$$IX = \sqrt{[(\%\ \text{Impedance drop})^2 - (IR)^2]}$$

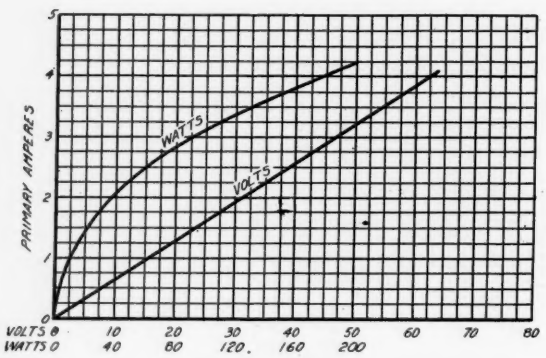


FIG. 10. CURVES FROM SHORT-CIRCUIT TEST OF TRANSFORMER

Impedance drop is taken from data obtained in the short-circuit test. $(IR)^2$ is obtained from formula A above.

(C) "In phase" component of voltage drop, S = IR (power factor) + IX (wattless factor).

Wattless factor is obtained from the following:

When P. F. is	1.00	.90	.85	.80	.70	.60
Wattless factor is	.00	.42	.52	.60	.70	.80

"Right angled" component of voltage drop;

(D) t = IX (power factor) — IR (wattless factor)

(E) Then per cent regulation = S + ($t^2 \div 200$). At 100 per cent power factor the wattless factor is zero, so formula (E) becomes,

(F) Per cent regulation = IR + ($IX^2 \div 200$).

Formulas E and F are not absolutely correct, but they are a convenient means of solving for regulation and introduce an almost negligible error. For derivation the reader is referred to almost any standard text book.

Efficiency

$$\text{Efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{output}}{\text{output} + \text{losses}}.$$

It is usually calculated at ¼, ½, ¾, full and 1¼ load. In transformers which operate on a 24-hr. load, the figures for these various points are valuable in determining the desirability of a given type; in lighting service, however, there is a long period of no load which accentuates the no-load losses. For this kind of service the determination of what is known as "all day efficiency" is more important than the derivation of "load efficiency." "All day efficiency" is the ratio of $\frac{\text{integrated output}}{\text{integrated input}}$ during a 24-hr. period.

Example of All Day Efficiency

To make clear in a concrete way the various curves and tests above enumerated and described under internal losses, an actual case will be useful.

Name plate data 2200/220-110 volts; 60 cycles, single-phase high-efficiency type transformer, 7.5 kv.a. rating. From this we may calculate the following:

Ratio 10:1. Primary current $\frac{7500}{2200} = 3.41$ amp.

Secondary current $\frac{7500}{220} = 34.1$ amp.

The transformer has been standing in a room of 28 deg. centigrade for several days so that this may be safely taken as the coil temperature. The results of resistance tests as tabulated in Table III show resistance of primary and secondary to be 4.30 ohms and 1.0615 ohms respectively at 28 deg. C., which being reduced to resistance at 25 deg. becomes (using formula under "resistance"),

$$\text{Primary resistance at 25 deg.} = 4.3\left(\frac{263}{238+28}\right) =$$

4.25 ohms.

$$\text{Secondary resistance at 25 deg.} = 0.0615\left(\frac{263}{238+28}\right)$$

= 0.0608 ohms.

The copper loss by I^2R method now becomes as in Table V.

Table V. Copper Loss Table

Per cent Load	Primary Amps	Primary Ohms	Primary Watts	Secondary Amps	Secondary Ohms	Secondary Watts	Total Watts
25	0.85	4.25	3.0	8.5	.0608	4.4	7.4
52	1.70	4.25	12.3	17.0	.0608	17.6	29.9
75	2.55	4.25	27.6	25.5	.0608	39.5	67.1
100	3.41	4.25	49.5	34.1	.0608	70.8	120.3
125	4.26	4.25	77.0	42.6	.0608	110.0	187.0

It will be noted that the total watts above are less than the watts obtained on the short-circuit test and tabulated in Table IV. Figure 10 shows curves plotted from the short-circuit test.

The core loss and saturation test results give a set of curves as shown in Fig. 11. The data are those in Table II.

From the data above given the efficiency can be derived as follows:

TABLE VI. CALCULATIONS OF EFFICIENCY

	Full Load	½ Load	¼ Load
I^2R Loss	120.3	29.9	7.4
Core loss	60.0	60.0	60.0
Total loss	180.3	89.9	67.4
Output	7500.0	3750.0	1875.0
Input	7680.3	3839.9	1942.4
Efficiency = Per cent	$\frac{7500}{7680} = 97.7$	$\frac{3750}{3839.9} = 97.6$	$\frac{1875.0}{1942} = 96.7$

Some authorities use copper loss as taken from Table IV, thus allowing for eddy current losses. As a comparison, full-load efficiency based on these figures is shown below.

I^2R + load loss	127
Core loss	60
Total	187
Output	7500
Input	7687

$$\text{Efficiency} \quad \frac{7500}{7687} = 97.6 \text{ per cent}$$

The effect of this method of figuring is almost negligible.

All-day efficiency uses the standard assumption that during 24 hr. the transformer operates 5 hr. at full load and 19 hr. at no load.

Output = 5 hr. at 7.5 kw. = 37500 watts
Output = 19 hr. at 0 kw. = 0 watts

37500 watts

Input 5 hr. at 7680 w. = 38400 watts
19 hr. at no load = (19 × 60 watts) = 1140 watts

39540 watts

$$\text{All-day efficiency} = \frac{37500}{39540} = 95 \text{ per cent.}$$

For purposes of comparison, it is interesting to note the case of a transformer of the same full-load efficiency, but having 60 watts I^2R loss and 120 watts core loss.

Output as before 37500
Input, 5 hr. at 7680 = 38400 watts
19 hr. at no load = (19 × 120) = 2280

40680

$$\text{Efficiency} \quad \frac{37500}{40680} = 92.5 \text{ per cent.}$$

This illustrates the necessity of what is technically known as "balanced design."

The full load regulation at 100 per cent power factor and at 80 per cent may be determined as follows, using formulas previously given:

Primary IR = 4.25 × 3.41 = 14.5 volts
Secondary IR = 0.0608 × 34.10 = 2.07 volts

Then from (A),

$$\text{IR per cent} = \left(\frac{14.5}{10} \times 2.07\right) \div \frac{3.52}{220} = 1.6 \text{ per cent}$$

Impedance drop (Table IV) = 54 volts = 54 ÷ 2200 = 2.45 per cent.

Then from (B), $IX = \sqrt{(2.45^2 - 1.6^2)} = \sqrt{(6 - 2.56)} = \sqrt{3.44} = 1.85$ per cent = reactive drop.

At 100 per cent P. F. wattless factor = 0. Hence S = IR and t = IX.

From (F) per cent regulation = 1.6 + (1.85² ÷ 200) = 1.6 + 0.0172 = 1.61 per cent = regulation.

At 80 per cent P. F., wattless factor = 0.60.

S = (1.6 × .80) + (1.85 × .60) = 2.39
t = (1.85 × .80) − (1.60 × .60) = .52

From (C) and (D), per cent regulation = 2.39 + (0.52² ÷ 200) = 2.39 + 0.0013 = 2.391 per cent regulation.

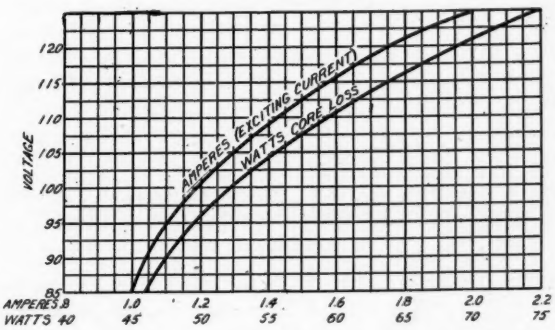


FIG. 11. CURVES FROM CORE-LOSS TEST

If, for any reason, it is necessary to determine the rate of transformation of a transformer, the simplest method is to read simultaneously on two voltmeters, or read on one voltmeter equipped with a double-throw switch, the voltage on the primary and secondary. In making this test, it will usually be necessary to have a standard potential transformer to use in connection with the voltmeter in reading the high-voltage side.

In order to determine whether or not the transformer has its leads brought out in accordance with Fig. 1, the following simple test is used: Connect as shown in Fig. 12 and measure voltage from A to B and from A to C. If A to B gives the higher indication, the leads are brought out incorrectly; but, if A to C gives the higher reading, the connections are correct, and are schematically arranged as in Fig. 1.

Tests above described are those usually performed and are sufficient to indicate accurately the characteristics of a given transformer.

EVALUATION OF LOSSES

IN ORDER to use intelligently the figures obtained, it is necessary to evaluate the losses in terms of dollars and cents. An example of a more or less rough and ready

determination will show how this is done. Assume a lighting transformer of a type similar to the one which has been under consideration. Tests show it to have an all-day efficiency of 91 per cent, i. e., its input is 41,200 watts, an increase of 1640 watts over the high efficiency type per 24 hr. or about 600 kw.-hr. per year. If the cost per kw.-hr., distributed, is 2 cents, this would mean a yearly saving of $12 as compared with a standard high-efficiency transformer, without allowing anything for station capacity cost. At present prices a 7½-kv.a. transformer costs $108; the old transformer is worth about $15, making a net cost of $93. Including the cost of making the change, etc., the total investment would be about $100, on which the central station receives $12 as interest and depreciation.

The consequential losses following upon poor regulation are various, but perhaps the most striking example is that furnished by the variation of wattage of lamps under varying voltage. For example, if a 110-volt lamp is operated on a 105-volt circuit, it consumes only 92.9 per cent of its rated wattage; and what is more important, gives off only 84.9 per cent normal candle power. Thus a very small decrease in voltage affects seriously both the central station's income and the service rendered to consumers.

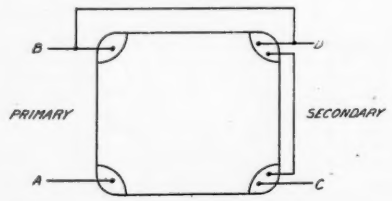


FIG. 12. METHOD TO TEST TRANSFORMER LEAD CONNECTIONS. CONNECT TESTING CURRENT TO A AND B. THEN CONNECT B TO D

Characteristic Performance Data

As an aid in checking efficiencies, the following Table VII shows the data on a high grade line of transformers for distributing service.

TABLE VII. DATA OF TRANSFORMERS

Rating k.w.	Core Loss Watts	Copper Loss Watts	Regulation Per cent	Efficiency % 100% Load	Efficiency % 50% Load
1	19.5	26.5	2.75	95.7	95
3	34	62	2.16	97	96.98
5	44	88	1.86	97.39	97.37
10	79	144	1.58	97.8	97.73
20	132	297	1.54	98	98.03
30	200	350	1.26	98.24	98.12
50	270	565	1.22	98.35	98.36
75	385	700	1.00	98.60	98.50
100	520	880	.95	98.6	98.5*
200	1450	1700	.95	98.4	98.1*

*For a different design.

It will be noted that efficiencies improve in the larger sizes. Years ago it was customary to install a 0.6-kv.a. transformer in front of every house, but today secondary networks are used, each network being fed by several larger transformers, 7½, 10 or even 25 kv.a. if in a congested district. The advantages of this are, first, in the higher efficiency of the larger transformers; and, second, and of much more importance, the saving in investment and in losses due to taking advantage of the diversity factor of a residence load. Under the old scheme, a house with 600 watts connected load had to have a 600-watt supply even though it was fully loaded only once a year. Under the new scheme, 10 houses with a total connected load of 6000 watts can be carried on a 2-kw., for the simple reason that seldom, if ever, do more than two or three of the houses have their maximum load on simultaneously.

In laying out a distribution system, every effort must be made (consistent with good service) to limit the number and kv.a. of transformers installed to the very minimum, thus reducing materially the all-day losses. Similarly in a substation, where proper switching facilities are provided, it is economy to cut out unnecessary banks of transformers during off-peak periods.

Isolated plants having "standby" service from a central station can save energy and coal for themselves or the station by providing oil switches in the primaries for eliminating core loss during the long periods of idleness.

Checking Distribution Systems

In a distribution system consisting of a large number of transformers, it often happens that transformers are improperly loaded. It is not at all uncommon to find a 10-kv.a. installation carrying a 2-kv.a. peak, and a 5-kv.a. carrying a 10-kv.a. peak. Such errors are due to shifting loads and can be eliminated by means of a careful and thorough survey.

The quickest method of making this is to equip a "flying squad" of men with split core current transformers (such as are built by the General Electric and Westinghouse Cos.) and ammeters. Long leads are provided so that the lineman can clamp the split core around the secondary leads of a transformer under test, while the assistant on the ground reads the meter. With a car, two men can cover from 20 to 30 transformers in a night, readings being taken, of course, only during peak hours. Early fall is the most favorable time for this work.

A slower but more accurate method is to substitute for the indicating ammeter a recording instrument, leaving this on each transformer 24 hr. This method gives an accurate curve of load variations, but on large systems consumes a great deal of time.

Such a survey, carefully made, and intelligently acted upon, will materially reduce no-load losses, and in addition permit of a re-arrangement of apparatus which may save large sums of money by making new purchases unnecessary. One of the commonest results of a survey is a store room full of extra transformers. If these are old and non-standard, they should be given a series of tests as described in this article, and if found to be inefficient or unsafe, discarded. Newer types should be given a "puncture test" with high voltage, and if time permits, a core loss test. Any ingenious mechanic can improvise a test table which by making con-

nections convenient will reduce the time of a given test materially.

Resume

Be sure that transformers are connected properly and are of the proper size and ratio.

Test insulation as often as a transformer comes into the storeroom.

Check old types of transformers and discard all having high losses.

Examine oil from time to time, and filter when necessary.

A conscientious observation of these precautions can not fail to increase efficiency of operation and reduce the number of burn-outs, to say nothing of conserving human life.

Electric Motor Tests

By Ottomar H. Henschel

INSTRUCTIONS given relative to the inspection of electric generators, pages 54 to 58, apply equally well to motors. If machine windings have been damaged or subjected to moisture, insulation resistance tests are advised. Measurements may be made by using 500-v. direct-current, and a 500-v. direct-current voltmeter, employing the scheme of connections indicated in Fig. 1. Read the line voltage, connect resistance to be measured in series with voltmeter, and take a second reading.

Dividing the product of the resistance of the voltmeter winding, in ohms, and the difference between line voltage and the voltage reading with insulation in series with voltmeter by one million times the voltage reading with insulation in series with voltmeter, will give as a quotient the resistance of insulation in megohms (one million ohms).

If grounded circuit is employed, connect grounded side of line to frame of machine and the voltmeter between the windings and other side of circuit.

Insulation resistance should be approximately one megohm for each 10,000 v. applied in test, and no machine should have an insulation resistance of less than one megohm.

Motor speed tests may be conducted by measuring, under loads ranging from zero to 150 or 200 per cent of machine rating, the number of revolutions per minute of the revolving member employing either some method of count, or some form of speed indicator. Curves obtained by plotting load values against corresponding speed values indicate the performance of the machine.

Due to increasing field strength, the speed of a series-wound motor will decrease with increasing load in the manner indicated in Fig. 2, while with a shunt-wound, direct-current machine, in which the field strength remains practically constant throughout a considerable range of load, drop in speed from no load to full load seldom exceeds 5 per cent. Figure 2 also illustrates a typical compound-wound motor speed curve, the dropping of which may be accounted for by the effect of the increasing strength of the series field upon the constant field created by the shunt winding. In a differential compound-wound motor, the series winding opposes the shunt winding and, as a consequence, the speed, if not remaining constant, increases with load.

Speed curves of various types of alternating-current motors are shown in Fig. 3. Throughout its rated capacity, the induction motor may be regarded as a constant speed machine, its change in speed from no load to rated load varying from 4 to 8 per cent, depending upon the capacity, the larger sizes usually having the better degree of regulation.

Efficiency

Efficiency of machine is expressed as the ratio between the output and the input, the latter in the case of a motor being measured in the same manner as the output of a generator. Method of motor output measurement depends upon type of drive employed. If belted, the number of horsepower transmitted may be approximated by dividing the product of the velocity of the belt, in feet per minute, and the width of the belt, in inches, by 550, while if direct drive is utilized, other means must be resorted to.

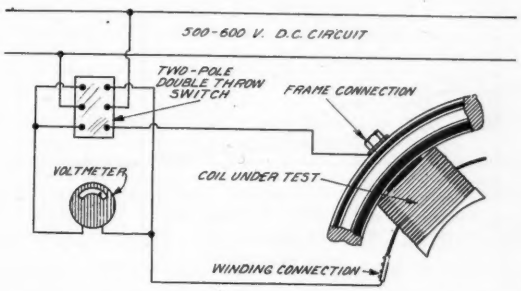


FIG. 1. DIAGRAM OF CONNECTIONS FOR INSULATION RESISTANCE TESTS

Measure electrical energy requirements under various load conditions. Fit the motor shaft with a pulley having a diameter the same as the pitch diameter of the pinion or gear removed, and determine the number of horsepower developed by application of a Prony brake arranged in the manner indicated in Fig. 4. Adjust the load as desired and note speed of revolving member, in revolutions per minute, and reading of scale when in a state of balance. Dividing the product of horizontal distance L from center of shaft to point of application on scale platform, scale reading in pounds, number of revolutions per minute of rotating member and 6.2832, by 33,000 will give number of horsepower developed.

Efficiencies of various types and sizes of direct and alternating-current motors are given in the tables, Figs. 5 and 6, respectively.

Curves and data furnished by motor manufacturers enable one to determine the efficiency of a motor at a given load. With machine running under normal voltage, measure current flow by means of an ammeter con-

nected in the supply circuit and calculate per cent load being carried by multiplying by 100 the quotient obtained by dividing the ammeter reading by full load rated ampere capacity of the motor as given on the name plate. Reference to manufacturer's curve will give corresponding efficiency.

Tests of this nature are preferably made under maximum load conditions, and if found that the motor is not operating, for the greater part of the time under highest degree of efficiency obtainable, a machine of different size should be substituted in the case of individual drive or rearrangement of driven machines made where group drive is employed. Make only such changes as will insure working the motor under a load which corresponds as nearly as possible to maximum efficiency.

Capacity

Where motors are made to carry momentary heavy overloads, but otherwise are running considerably under-

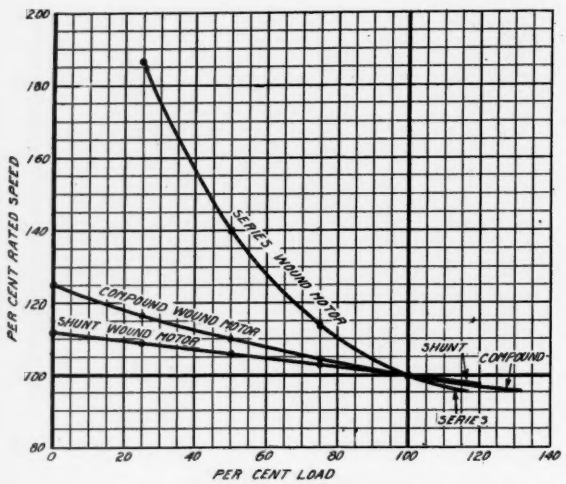


FIG. 2. TYPICAL DIRECT-CURRENT MOTOR SPEED CHARACTERISTICS

load and at exceedingly low efficiency, and in the case of alternating-current induction motors at low power factor, smaller machines equipped with sufficiently heavy flywheels may often be used instead to advantage. Under such conditions, employ a motor of such size as will most readily carry the average load. The size needed may be obtained from the records of a recording ammeter connected in series with the machine, although where such an instrument is not available, the same result may be secured, at least approximately, by the average of a series of indicating ammeter readings taken every minute over a given period of time.

Where new machines or groups are being installed, marked savings may be realized by determining the actual requirements before purchasing the driving motor. Temporarily install a motor of size sufficient to operate the machine or group under test, determine the actual power requirements from instruments connected in the supply circuit in the manner indicated above. From the results obtained (in watts) subtract the losses occurring within the motor at the average load under which it operates and divide the difference in watts by 746 to obtain the number of horsepower necessary to drive the machine or the group as the case may be.

The motor used for this purpose may be temporarily installed at the point it is desired to place the regular machine or, as in the case of large installations where frequent changes are made in load and arrangement of driven machines and a motor is kept in stock only for

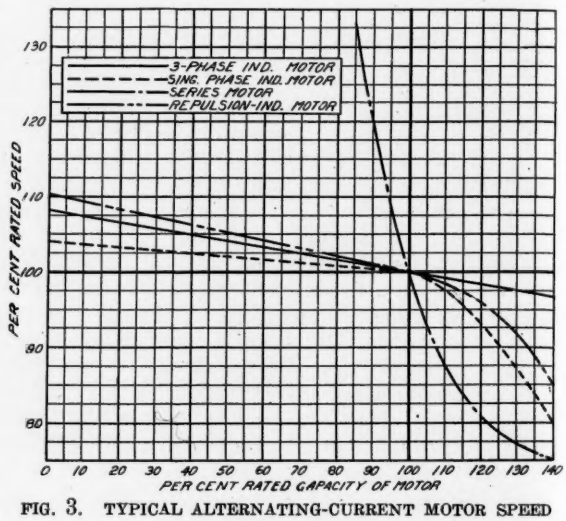


FIG. 3. TYPICAL ALTERNATING-CURRENT MOTOR SPEED CHARACTERISTICS

such test purposes, the mounting shown in Fig. 7 may be utilized to good advantage, equipping the apparatus with a counter shaft and a set of cone pulleys so as to have available a comparatively large range of speeds.

A scheme of electrical connections, particularly convenient for checking up motors already in place, is illustrated at A and B, Fig. 8. In the case of a direct-current or a single-phase alternating motor, a "blown" fuse fitted with dummy connectors tied in with the ammeter or the current coil of the wattmeter is used instead of the regular fuse, thus eliminating the necessity

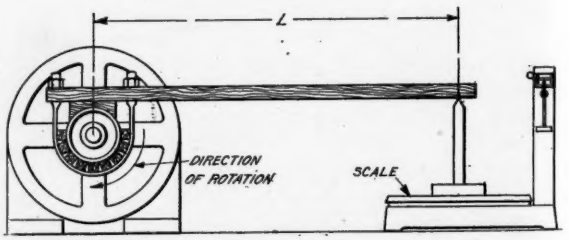


FIG. 4. ARRANGEMENT OF EQUIPMENT FOR A BRAKE TEST

of interfering with any permanent connections. With a three-phase machine, two of these dummy fuses will have to be used, and in either case, in order to protect the machine under test fully, the circuit leading to the measuring instrument should be fitted with a portable fuse of capacity equal to that removed from the fuse block.

A single-pole knife switch mounted on the dummy fuse in the manner shown in Fig. 9 will allow cutting

out the ammeter or the current coil of the wattmeter without interfering with the operation of the motor.

Starting Requirements

Power required to start a machine or group of machines may, in the case of belt drive, be determined by measurement of the torque in the manner indicated in Fig. 10. With a spring scale attached to the belt by means of a wooden clamp and held parallel to the belt,

FIG. 5. EFFICIENCIES OF VARIOUS TYPES AND SIZES OF DIRECT-CURRENT MOTORS

Horsepower Capacity	Efficiency			Horsepower Capacity	Efficiency		
	1/2 Load	3/4 Load	Full Load		1/2 Load	3/4 Load	Full Load
7.5	77	81	82.5	135	89	90.5	91
15	82	85	86	165	90.5	91	91
20	82.5	86.5	86.5	200	90.5	91.3	91.5
25	84	86.5	87.5	265	91	91.5	92
35	85	88	89	400	91.3	91.8	92
45	87	89	89.5	500	91.8	92.3	92.5
65	88	89.5	90.5	650	91.8	92.2	92.5
80	88.5	90.5	91	1000	92	92.3	92.5
100	88.5	90.5	91	1250	92.5	93	93.5
120	88.5	90.5	91				

the motor is started sufficiently to turn the driven pulley a small part of a revolution, and the scale reading noted. This is the force in pounds required to start the load at a radius of L ft.; the torque in pounds-feet is equal to the product of this force in pounds and L in feet, while the value of the number of horsepower required is equal to the quotient obtained by dividing the product of the torque in pounds-feet and the speed of the driven pulley in number of revolutions per minute by 5252.

Where a direct-connecting coupling is employed, the scheme indicated at B, Fig. 10, may be utilized, while if

FIG. 6. EFFICIENCIES OF VARIOUS TYPES AND SIZES OF ALTERNATING-CURRENT MOTORS

Horsepower Rating	Number of Poles	Single Phase				Polyphase			
		1/2 Load	3/4 Load	Full Load	5/4 Load	1/2 Load	3/4 Load	Full Load	5/4 Load
1	4	60	63	68	62	74	78	79	79
2	4	71	75	78	77	77	81	82	82
5	4	71	76	77	76	82	84	85	84
10	6	75	79	80	79	84	85	85	83
20	6	85	88	86	85	85	87	87	86
30	8	77	81	83	82	87	89	88	87
50	8	82	84	86	86	87	89	90	90
75	10	--	--	--	--	88	90	90	89
100	10	--	--	--	--	88	90	90	90
150	10	--	--	--	--	88	90	91	90

the motor shaft is fitted with a gear, the scale may be attached directly to one of the arms of the gear as at C, Fig. 10.

Slip Tests

Induction motors may be tested for slip by attaching to the shaft of the machine a black disk with p white radial lines or sectors painted upon it, where p is the number of poles of the motor. If, with the motor running, the disk is illuminated by means of an alternating-current arc lamp connected to the same lines as the motor, synchronism is indicated by the stationary appearance of the white lines on the disk. If, however, the motor is below synchronism, the disk will appear to rotate at a speed equal to the difference between the

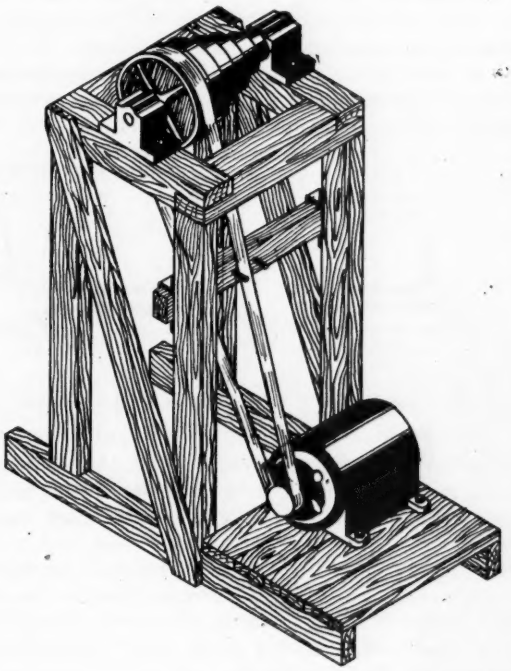


FIG. 7. CONSTRUCTION OF TEST MOTOR RACK FITTED WITH CONE PULLEY

synchronous and the actual speed. Dividing the number of revolutions at which the disk seems to rotate by the synchronous speed of the machine, and multiplying the quotient obtained by 100 will give, as a result, the per cent slip.

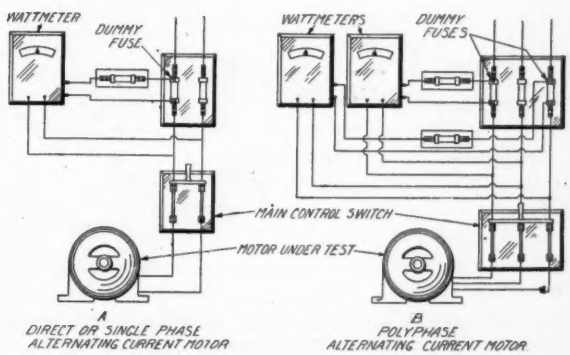


FIG. 8. DIAGRAMS OF CONNECTIONS FOR CHECKING UP MOTOR LOADS

In a reasonably good motor the per cent slip should not be in excess of 12.

Predetermining Performance of Induction Motors

With the aid of the total no load watts, W_0; the no-load amperes, i (per phase); the total (short circuit)

watts, W_s, with rotor blocked, and the total short circuit current i_s (per phase) all at rated machine voltage E, a circle diagram may be constructed by means of which the performance of a polyphase induction motor may be predetermined.

As indicated in Fig. 11, lay out on suitable co-ordinate paper axis OE and OX, the former representing the value of the impressed electromotive force. To a convenient scale take OP equal to 100 per cent power factor and subdivide as shown. With OP as a radius describe quadrant PRQ, the point of intersection of which with

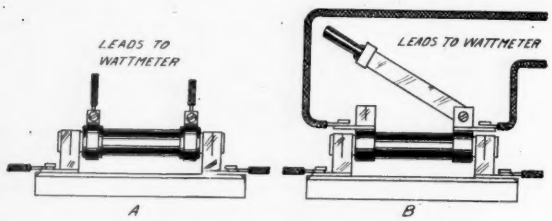


FIG. 9. METHODS OF CONNECTING DUMMY FUSES AS EMPLOYED IN FIG. 8

the primary current vector when extended horizontally to the vertical scale OP gives the power factor reading corresponding to the location of the primary current vector.

With rotor blocked, apparent horsepower input is equal to the quotient obtained by dividing $\sqrt{3}\ i_s\ E$ by 746. Plot this value as vector OI_s, the angle with OE being dependent upon the power factor determined by dividing 100 W_s by $\sqrt{3}\ i_s\ E$.

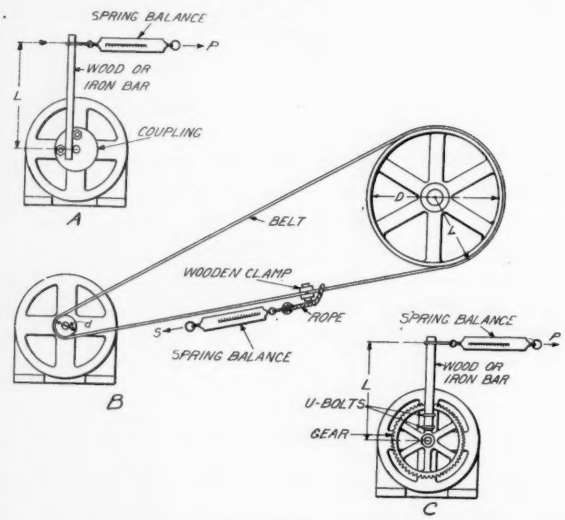


FIG. 10. METHODS EMPLOYED TO MEASURE MOTOR TORQUE

OI_0 represents apparent horsepower input at no load and is equal to $\sqrt{3}\ i_0\ E$ divided by 746, the power factor being obtained by dividing 100 w_0 by $\sqrt{3}\ i_0\ E$.

Draw I_0 M the power component; O M is the wattless component.

Through I_0 and parallel to OX draw I_0K, using this as a base for the semi-circle drawn through points I_0 and I_s and representing the path of the current as the load on the motor increases. Vector OI drawn for a given load will represent the apparent horsepower input, while D measures to the same scale the true horsepower input.

OI may also represent the value of the primary current, and is reducible to this by dividing 746 OI by $E\sqrt{3}$.

Joining I_0 and I_s will give the output as represented by IA, thus rendering it possible to determine the effi-

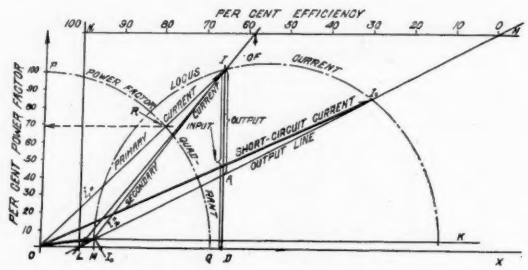


FIG. 11. TYPICAL CIRCLE DIAGRAM FOR PREDETERMINATION OF INDUCTION MOTOR PERFORMANCE

ciency by dividing IA by ID, although this may be obtained more directly by construction of the efficiency scale. Produce I_0 I_s to L and erect at that point perpendicular LN_1, N_1, at such a distance from OX as will allow the ready division of NN_1, into 100 equal parts.

To determine per cent efficiency corresponding to input ID, draw LI extended to the efficiency scale, the point of intersection giving the per cent efficiency.

The power factor for any given value of primary current as OI may be obtained by projecting horizontally to the power factor scale the point of intersection of OI with quadrant PRQ.

FIG. 12. TABLE OF TEMPERATURE COEFFICIENTS FOR USE IN CALCULATING MACHINE TEMPERATURE RISE

t Initial temperature, Centigrade	a Temperature coefficient	t Initial temperature, Centigrade	a Temperature coefficient
0	0.00420	26	0.00379
1	0.00418	27	0.00377
2	0.00417	28	0.00376
3	0.00415	29	0.00374
4	0.00413	30	0.00373
5	0.00411	31	0.00372
6	0.00410	32	0.00370
7	0.00408	33	0.00369
8	0.00406	34	0.00368
9	0.00405	35	0.00366
10	0.00403	36	0.00365
11	0.00402	37	0.00364
12	0.00400	38	0.00362
13	0.00398	39	0.00361
14	0.00397	40	0.00360
15	0.00395	41	0.00358
16	0.00394	42	0.00357
17	0.00392	43	0.00356
18	0.00391	44	0.00355
19	0.00389	45	0.00353
20	0.00388	46	0.00352
21	0.00386	47	0.00351
22	0.00385	48	0.00350
23	0.00383	49	0.00348
24	0.00382	50	0.00347
25	0.00381		

Temperature Determinations

Machine temperatures may be obtained by means of thermometers applied to the hottest accessible part, by increase of winding resistance and by the imbedded temperature-detector method. With thermometers the actual temperature is estimated by adding to the highest observed reading a correction factor of 15 deg. C., except where the instrument is applied directly to the surface of a bare winding in which case but 5 deg. C. is added.

In employing the measurement of resistance method, temperature rise in degrees Centigrade is obtained by dividing the difference between the resistance of the windings before test and their resistance at time of shutdown, by the product of the resistance at time of shutdown and the temperature coefficient, the value of which may be obtained by reference to the table shown in Fig. 12. For ordinary purposes, an initial temperature of 25 deg. C. (77 deg. F.) may be assumed, in which event a temperature coefficient of 0.003,88 is used.

In the case of two-layer windings having thermocouples between coils and between coil and slot, add 5 deg. C. to the highest reading. In single-layer windings with detectors between coil and core, and between coil and wedge, add to the highest reading 10 deg. C., plus 1 deg. C. for each 1000 v. above 5000 v. terminal pressure.

Leather Belting

Choosing and Caring for Belts, Proper Inspection, Proper Tension, Arrangement of Drives. By Claude O. Streeter

NO ONE article used in the mechanical world figures so conspicuously in the operation of machinery as leather belting, and likewise there is no one article which is so little understood.

Unfortunately with most people who buy it, a belt is a belt, the price, not the quality, of the belt being to a great extent, the deciding factor. A belt of the same ply and thickness is used for all purposes; then when the belt fails, no matter under what conditions it may have been used, it is condemned.

As a matter of fact, vastly more leather belts fail from improper mechanical conditions and lack of care, than from being made of poor material.

This subject of Leather Belting is very broad, and this article aims to endeavor to offer a few helpful suggestions only on the principal points of this subject.

Selection of Leather Belting

In selecting leather belting, it is most important to consider quality. Failure to do so will result in poor service, short belt life, and unnecessary expense caused by the necessity of purchasing two or more belts, when one properly chosen would have outlasted them.

This does not mean that it is necessary to purchase the most expensive belt in all cases, but to purchase belts with a regard for mechanical conditions of the drive on which they are to operate.

The points to be observed about a drive before selecting the belt for it are diameters, faces and speeds of both pulleys; distance between pulley centers; type of drive; physical conditions surrounding the drive; the horsepower requirements of the drive.

In connection with these observations, it is well to remember the following facts:

Pulley diameters control belt thickness or weight to be used.

Belt speed controls belt power transmission.

Power requirement controls belt width.

The physical conditions will inform one as to whether or not a waterproof or special tannage belt is required.

Another point to remember in selecting a leather belt, is that all leather belts are constructed on a basis of light, medium and heavy belts.

Light belts are approximately 1/8 in. thick in single and 1/4 in. thick in doubles. Medium belts are approximately 5/32 in. thick in single and 5/16 in. thick in doubles. Heavy belts are approximately 3/16 in. thick in single and 3/8 in. thick in doubles.

Now fix in the mind the comparative thickness of each weight and we are able to imagine the various thicknesses of belt bending around a pulley of given diameter, and to judge intelligently as to the advisability of installing a heavy double, 3/8 in. thick, or a light double belt, 1/4 in. thick, on a pulley of 7 in. diameter, for instance. Your selection will certainly be the light double belt. In order to get the effect of bending belts, of the above thickness, clearly fixed in mind, it might be advisable to procure sample pieces with which to experiment.

With a careful observance of the various facts mentioned above, together with a working knowledge of leather belting formulas, one is in a position to select the proper belt for any drive. From these statements it will be observed that the basis for belting selection is really thought, close observation, and a sense of proportion.

In selecting a leather belt of proper weight, or thickness, for a pulley of given diameter, the following might be considered as a guide.

	Single Belts	Double Belts
Light	4 in. diam. pulley	7 in. diam. pulley
Medium	7 in. " "	12 in. " "
Heavy	10 in. " "	15 in. " "

It must be kept in mind that the above is merely a guide for the proper diameter pulleys to be used for best results with an oak tanned belt of given thickness (a special tannage will operate on smaller pulleys) and not as the smallest diameter which can be used with these belts; for, while it is not good practice, it is nevertheless a fact that many times on slow drives a belt is merely thrown over a 1 15/16-in. shaft and no pulley whatever used.

If a change is to be made in the above diameters, it is always advisable to increase rather than decrease them.

Power Transmitting Capacity of Leather Belting

It seems to be the general custom when writing on this particular phase of leather belting, to deluge the reader with formulas and charts so that he does not know nearly as much about this point after reading as he did before.

While it is necessary to have a working knowledge of belting formulas, it is not the purpose of this article to give a complete discourse on belting and its formulas, but a few practical hints as an aid to understanding them. Therefore, there will be no formulas in this article.

There are in reality only a few simple formulas connected with the use of leather belting, and these can be secured in belting manuals.

Under Selection of Leather Belting, the bearing of speed and width on the transmitting capacity was defined. Therefore in order to estimate the power transmitting capacity of a leather belt from its width and speed, it is only necessary to remember three values as follows:

A single belt traveling 600 ft. per min.; or a double belt at 375 ft. per min.; or a triple belt at 300 ft. per min. will transmit 1 hp. for every inch of width.

These figures are based on the assumption that both pulleys are of approximately the same diameters. From the above it will be observed that by knowing the speed and width of a belt, its power capacity can easily be estimated.

Care of Leather Belting

Having selected and installed the proper belting throughout a plant, it is wrong to assume that the belting should be given no further attention, and thought. Belts as well as machines require attention. No one would think of starting a plant without a competent man to take charge of, and repair the machines. Why, then, expect the belting on which these machines are directly dependent to operate with little or no attention except that of an unskilled man?

A few suggestions on this point may be helpful:

Check up your shafting and machines periodically to see if everything is properly in line. As much belting is ruined by improperly lined shafting and pulleys as in any other way.

Belts on drives in this condition are often kept on the pulleys with guides placed beside the drive, or rub against shipper fingers, hangers, pulley flanges or part of machines, until a lap opens and the belt is torn or until it stretches on one edge. Belts once stretched in this way will not run straight, and are apt to run off the pulley, ruining themselves, and anything else which may be in their way.

Remember that because shafting was in line once, it does not necessarily remain so indefinitely and regular attention to this point will result in an economy not only in belting but in power.

Inspection of Belting

Place the belting in charge of one man and hold him responsible for it. Allow no belting installations, repairs, alterations or dressing except by him or under his direction.

It is of course desirable to put an experienced belt man on the job, but where this is not practicable a man should be selected and given a short course of instruction in a belt factory, so that he may learn how to make laps, take up belts, repair and dress belts, to work up old belts, etc. The principal object is to get a man on the job who is big and intelligent enough to run it economically; to put in the proverbial stitch in time and not wait until there is a breakdown causing a shutdown and delay and possibly serious accident. Remember that "Anybody's Job is Nobody's Job" and pin this proposition down.

It should be the duty of the belt man to inspect all belts periodically and ascertain which of the following conditions exist:

Belt all right in every respect.
Belt too dry.
Belt too oily.
Belt too heavily dressed.
Belt laps started to unglue.
Belt plies started to unglue.
Belt too slack or too tight.
Belt running true on pulleys.
Belt lacing, good or bad.
General condition of belt and drive.

The above is merely a suggestion as to the points for observation. If there is anything wrong about the belt it should be repaired at once and the belt put into first class condition or replaced, whichever is necessary.

The work done should be carefully reported and entered on the Belt Record Card for the particular drive. This card should contain complete data on belt and drive with spaces for entering work done on either so that the history and condition of both or either of them is available at a glance.

The above suggestions need not be expensive to follow, and are just as vital to the small shop as to the large.

Treatment of Leather Belting

Where a belt appears harsh and dry it is an indication that the natural oil has been exhausted from it and needs replenishing. The belt should first be carefully cleaned of surface dirt. This can usually be accomplished by rubbing with a piece of burlap or other coarse cloth. Naphtha or kerosene may be used to wash the belt if necessary. In extreme cases it may be necessary to use some sort of scraper; a sharp edged wooden block answers nicely. It is absolutely necessary to have a surface which can be recognized as leather on both sides of the belt, as it is useless to put dressing on a belt already covered with dressing, dirt or oil.

The best method of dressing a belt is to put a little on the pulley side and more on the outside of the belt, always remembering that too much dressing is worse than too little. Apply only what the belt readily absorbs. The pulley side of the belt should not be greasy when the application is completed. The best time to apply dressing is when the belt is idle and can stand

for some time so that the dressing can penetrate into the leather.

Avoid surface dressings of all kinds as they only collect dirt and cause the belt to slip. To remedy this slip more dressing is applied and so on until the belts acts only as a carrier for a sticky gum. The only excuse for the use of surface dressing is as first aid treatment in an emergency but it should be removed from the belt at the first opportunity.

Most users of belt dressing do more harm than good to their belting because they use too much dressing. Neatsfoot oil and castor oil make very good dressings; but the user invariably pours it on a belt from a can with little regard as to how much or how often he applies it, and in a short time his belt is so saturated with oil that it is like a rag and is materially weakened.

While the two oils mentioned make good dressings, they are apt not to be used successfully by the average mechanic and it is therefore undoubtedly best to use a commercial combination of greases as it is less subject to misuse and it is undoubtedly more economical.

The idea of dressing a belt is to make it soft and pliable and not oily or sticky.

Tension

There has been much discussion in the past few years in regard to the proper tension at which a belt should be installed and maintained. Special tools and instruments have been devised to make sure that the belts were installed at a predetermined tension, and much propaganda has been floated ridiculing the "old" practice of putting belts on by judgment, as far as tension is concerned.

Nevertheless, none of the schemes offered can be operated successfully without the use of considerable judgment, the tension being diminished in accordance with that judgment or experience.

For instance, a belt can be installed with a greater tension on a self-alining bearing than on a rigid pillow block bearing, and again a belt can be installed with greater tension on a 2 3/16-in. shaft than on a 15/16-in. shaft; yet belts of a given width are to be installed at a given tension without regard to mechanical conditions of the drive or the nature of its load, according to the tension idea of the theorists.

The best tension at which to install a belt is that at which the belt will carry its load easily without straining and with a minimum of journal friction. Operate all leather belts just as slack as possible without slipping. When it is necessary to operate a belt tight on a drive in order to make it pull, that is a sure indication that there is something wrong with that drive and it should be carefully investigated and the difficulty remedied.

With an intermittent load it is often necessary to install and maintain belts at a high tension.

When the belts are put on and taken up under the direction of one man, as advocated previously, the errors of too tight or too slack belts are avoided and a uniformity of belt tensions exists throughout the plant which can be obtained in no other way.

Care of belting may be summarized as follows:

Place the belting in charge of one man.
See that he knows his business and attends to it.
Have all belting inspected periodically.
Remedy all troubles in their beginning.
Keep all driving surfaces clean.
Put dressing into too dry belting.
Take grease out of too oily belting.
Run belts as slack as they will work efficiently.
Keep all shafting in line.

Types of Belt Drives

There are many different types of belt drives, but all of them are simply modifications of the three standard types known as open, crossed or quarter turn.

The ideal belt drive is an open horizontal, with the tight or pulling side on the bottom, but this ideal condition is not always possible, because of mill construction,

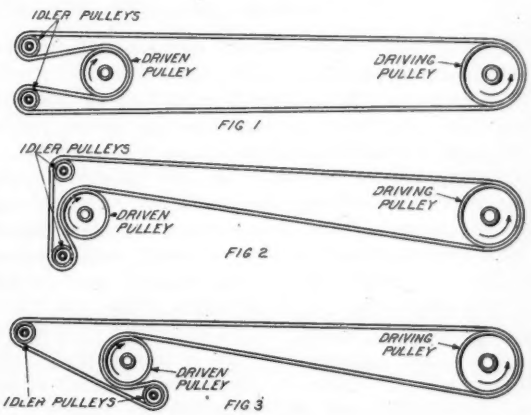


FIG. 1. USUAL ARRANGEMENT OF REVERSE DRIVE, PUTTING PULL ON IDLER. BEARING TROUBLE OFTEN RESULTS
FIG. 2. SUGGESTED REVERSE DRIVE WITH PULL ON DRIVEN PULLEY. VERTICAL IDLER ARRANGEMENT
FIG. 3. DIRECT-PULL REVERSE DRIVE WITH INCLINED IDLER ARRANGEMENT

shaft rotation, etc., and we are obliged to resort to other types of drives to meet these conditions.

Crossed drives are practical and satisfactory except when the difference in pulley diameters is too great, the centers short, or the power requirements call for a wide belt. Crossing a belt greater than 15 in. in width is poor practice, as it puts too great a strain on the belt edges.

When the above unsatisfactory conditions exist and it is necessary to reverse the rotation of the receiving pulley, we have recourse to the reverse drives shown in the accompanying sketches. Either or both of the idlers on all three of these figures can be made adjustable and used as binders to take up belt stretch.

Quarter turn drives are of four designs, two pulley, three pulley, four pulley and five pulley quarter turn.

The whole secret of success of this type of drive is in the proper alinement of the pulleys and most of the trouble experienced with this type of drive is caused by lack of proper consideration for this one item.

The most troublesome design of this type is the two-pulley quarter turn and much of this trouble is caused because of a lack of understanding of the drive and its

limitations. The pulleys on this design must be so placed that the point where the belt leaves the receiving pulley and the point where the belt leaves the driving pulley are in the same line. This drive can be operated in one direction only; that is, the drive cannot be run backward or reversed.

A leather belt of standard construction up to and including 5 in. in width will operate successfully on a two-pulley quarter turn drive, but when this width is passed, the belt must be specially constructed to meet the conditions of the drive under consideration.

Three-pulley quarter turn drives are merely a modification of the two-pulley type wherein an idler is brought into use to guide the belt or increase the contact arc on the pulleys. This drive properly laid out can be reversed and does not necessarily require belts of special construction.

Four or five-pulley quarter turn drives are the most satisfactory of any of the quarter turn designs, and when properly designed need no further attention than regular oiling of bearings and occasionally a taking up on the binder to allow for belt stretching. When possible this type should be used in preference to any other type of quarter turn. Of the last two mentioned designs the five pulley drive is preferable, especially on a vertical drive.

Compressing Air Efficiently

INSPECTION, CLEANING, EFFICIENCY TESTING AND REGULATION TO SECURE THE BEST RESULTS. BY CHARLES A. HIRSCHBERG

ECONOMICAL AND efficient operation of air compressors demands regular inspection. The compressor as a whole should be inspected at least once a month, making necessary adjustments, such as taking up of packing glands and replacement of worn parts.

Examine air valves, which should present an oily surface free from carbon. Also examine ports and passages for carbon or other obstructions. Carbon deposit on valves prevents their seating properly, causes leakage and results in reduced compressor capacity and low efficiency. Obstructions in air passages reduce their area and cause loss in friction. Carbon deposit in combination with high temperature due to improper cylinder cooling, sticking of valves or insufficient lubrication may result in explosions, if allowed to persist. Remove any carbon deposit.

PACKING

USE of packings is often misunderstood. Drawing up too tightly will cause heating, and scoring of the piston rod. This means higher temperatures and increased mechanical friction, with transmitting of increased temperature to the air in the cylinder, which will cause trouble with moisture, lower the air density and thereby reduce the output.

In placing the packing in the stuffing box, put the rings in so that the joints of the several rings stagger. Fill the box without crowding and with new packing force in slightly with the gland until the compressor warms up; then draw up just enough to prevent blowing. Crowding the packing will squeeze out the lubricant it contains, causing the packing to become hard and unyielding.

If upon examination packing is found hard, it may be revived by removing, loosening up, and working in a mixture of graphite and grease.

An asbestos sheet packing is generally used between cylinder heads and cylinders. If for any reason it becomes necessary to remove a head, care should be exercised not to injure this head gasket. If necessary to replace it, be sure to use material of the same thickness as the original, so as to maintain the clearance established by the builder.

INLET AIR

DO NOT have the air intake open to the engine room. Carry it outdoors and extend upward a sufficient height, with hood and screen, to prevent foreign substances from being drawn in and to insure cool intake air.

Figure 1 shows an ideal method of handling the air intake.

Take the air into the compressor as cool as possible is the rule without any exception. Cooling the intake air is a direct saving of power by reducing the volume to be compressed. Also the capacity of the air to hold moisture rapidly falls off as its temperature is reduced.

TRANSMISSION PIPING

PIPE lines should be of ample size, as small transmission lines mean excessive loss of pressure due to friction. Large pipe lines are especially desirable where the pipes are long or the supply has to meet the demands of a great many devices. Reduced pressure due to friction means inefficient operation of tools.

The accompanying chart, Fig. 2, covering "Loss in pressure due to friction in pipe lines" will serve to indicate the need for having the pipe line amply large. Take, for example, the following case:

The loss in pressure in transmitting 50 cu. ft. of free air per minute at 100 lb. pressure through 1000 ft. of 1-in. pipe is 11.89 lb., and only 0.27 lb. in transmitting the same volume of air the same distance through a 2-in. pipe.

Air lines should receive monthly inspection for leaks, which cost money and waste valuable compressor capacity. Assuming that 1 cu. ft. of air per min. at 100 lb. pressure costs $5 a year, this cost being computed from an estimated cost for electricity at 1c per kw.-hr., interest and depreciation at 10 per cent and a year as 300 8-hr. days, the following tabulation demonstrates the economy of regular pipe line inspections:

Diam. of hole, in.	Amount of Leak in cu. ft. free air per min.	Cost per year
1/16	6.45	$32
1/32	1.61	8
1/64	.40	2

Further calculations of possible loss through leaks may be quickly made by applying the above constant to the various discharge volumes shown in Fig. 3.

It is always advisable to provide numerous outlets in the transmission line. This will save future reconstruction and make it possible to use comparatively shorter lengths of hose. Where a number of supply lines radiate from the main line, it is well to provide a gate valve for cutting out any of these lines when they are to be idle for any considerable period of time.

Several methods are used of detecting air leaks: By the use of a lighted candle at all joints and connections; by swabbing soapy water around the joints, or by

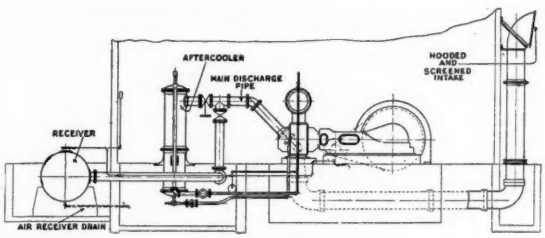


FIG. 1. PROPER ARRANGEMENT OF AIR INTAKE

following the practice of one Illinois manufacturer, who, at regular intervals, puts essence of peppermint into the air system and then has all the pipes inspected. Leaks are revealed by the odor.

Pipe lines may be tested for loss of pressure due to friction in transmission or to leakage by placing an air gage at the point of use of the air and comparing its reading with that of the gage on the air receiver. The remedies are increasing the size of the air line and eliminating of leaks.

Removing Moisture

All air contains moisture. When compressed the moisture becomes more evident, as in the compressed state there is more water in the air than it can contain. The excess moisture is deposited almost immediately as the air travels to the receiver; part of it travels with the air through the transmission lines, being gradually deposited and draining to low spots in the pipes, reducing the available area. Some of the moisture will find its way into the ports and exhaust passages of the machinery using the air, resulting in interference with proper operation by freezing at exhaust ports as the air expands to atmosphere.

One of the most effective and cheapest methods for the elimination of moisture from pipe lines is the placing of suitable traps at intervals in the transmission line into which the condensed moisture will precipitate and can be conveniently withdrawn.

A simple moisture trap, in Fig. 4, consists of an enlargement of the air line. It is provided with baffles and blowoff, the baffles having holes or slots drilled in the bottom so that moisture can work through to the blowoff pipe. By reason of the pressure always present in the separator the water can be taken from the top. This trap may be made of wrought or cast-iron pipe with flanged or screwed heads. A good proportion is separator diameter 3 times the air line diameter; 4 to 6 ft. long.

Another effective design is shown in Fig. 5, consisting of a long sweep tee three times the diameter of the pipe with a pocket below to catch the condensed moisture.

Both of these separators may be made locally from standard pipe fittings; also, there are a number of

FIG. 3. DISCHARGE OF AIR FROM ROUND HOLE IN RECEIVER, INNER EDGE ROUNDED, TO ATMOSPHERE

Receiver Gauge Pressure	Diameter of Orifice													
Pounds	1/16 inch	1/8 inch	3/16 inch	1/4 inch	5/16 inch	3/8 inch	1/2 inch	5/8 inch	3/4 inch	7/8 inch	1 inch	1 1/4 inch	1 1/2 inch	2 inch
	Discharge in Cubic Feet of Free Air per Minute													
2	.038	.153	.647	2.435	9.74	21.95	39.	61.	87.60	119.50	156.	242.	350.	625.
5	.0597	.242	.965	3.86	15.40	34.60	61.60	96.50	133.	189.	247.	384.	550.	985.
10	.0842	.342	.136	5.45	21.8	49.	87.	136.	196.	267.	350.	543.	780.	.....
15	.103	.418	1.67	6.65	26.70	60.	107.	167.	240.	326.	427.	665.	960.	.....
20	.119	.485	1.93	7.7	30.8	69.	123.	193.	277.	378.	494.	770.	.....	.....
25	.133	.54	2.16	8.6	34.5	77.	138.	216.	310.	422.	550.	860.	.....	.....
30	.156	.632	2.52	10.	40.	90.	161.	252.	362.	493.	645.	1000.	.....	.....
35	.173	.71	2.80	11.2	44.7	100.	179.	280.	400.	550.	715.	.....	.....	.....
40	.19	.77	3.07	12.27	49.09	110.45	196.35	306.80	441.79	601.32	785.40	.....	.....	.....
45	.208	.843	3.36	13.4	53.8	121.	215.	336.	482.	658.	860.	.....	.....	.....
50	.225	.914	3.64	14.50	58.2	130.	232.	364.	522.	710.	930.	.....	.....	.....
60	.26	1.05	4.2	16.8	67.	151.	268.	420.	604.	622.	.....	.....	.....	.....
70	.295	1.19	4.76	19.	76.	171.	304.	476.	685.	930.	.....	.....	.....	.....
80	.33	1.33	5.32	21.2	85.	191.	340.	532.	765.	1004.	.....	.....	.....	.....
90	.364	1.47	5.87	23.50	94.	211.	376.	587.	843.	.....	.....	.....	.....	.....
100	.40	1 61	6.45	25.8	103.	231.	412.	645.	925.	.....	.....	.....	.....	.....
125	.486	1 97	7 85	31 4	125.	282.	502.	785.	.....	.....	.....	.....	.....	.....

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FIG. 2. PRESSURE LOSS IN TRANSMITTING COMPRESSED AIR

Delivery in Cu. Ft. of Compressed Air per Min.	Equiv. Delivery in Cu. Ft. of Free Air per Min.	Size of Pipe, Inches: 1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	16
		Loss of Pressure, in Pounds, by Friction in 1000 Feet Lengths of Pipe																	
At 60 Pounds Gauge																			
9.84	50	18 24	5 06	1 95	.42	.13	.05												
14 73	75		11 34	4.33	.95	.29	.11	.05											
19.64	100		20.16	7 79	1.69	.52	.19	.08	.04										
24 60	125			12.23	2.65	.81	.30	.13	.07	.03									
29.45	150			17.53	3.80	1.16	.44	.19	.09	.05	.03								
34.44	175				5.17	1.58	.59	.26	.13	.07	.04	.01							
39.35	200				6.77	2.09	.78	.36	.17	.09	.06	.02	.01						
49.20	250				10.61	3.24	1.22	.55	.27	.15	.08	.03	.01						
58.90	300				15.20	4.65	1.78	.78	.38	.21	.12	.05	.02	.01					
68.6	350					6.31	2.37	1 07	.53	.29	.17	.06	.03	.01					
78.6	400					8.28	3.11	1 40	.69	.39	.22	.08	.04	.01					
88.4	450					10.47	3.94	1.77	.88	.48	.28	.11	.05	.02	.01				
98.4	500						4.88	2.20	1.08	.60	.34	.14	.06	.03	.01				
118.1	600						7.03	3.17	1.56	.87	.49	19	.09	.04	02	.01			
137.5	700						9.52	4.29	2.12	1.17	.67	.27	.12	.06	03	.02			
156.6	800							5.57	2.75	1.52	.87	.34	.15	.08	04	.03	.01		
176.5	900							7.08	3.49	1.94	1.17	.43	.19	.09	05	.03	.01		
196.4	1000							8.77	4.33	2.40	1.37	.54	.24	.12	06	.04	.02	.01	
294.5	1500								9.73	5.39	3.08	1.20	.55	.27	15	.09	.03	.01	
393.7	2000									9.65	5.51	2.16	.98	.41	.27	.16	.06	.03	.01
492	2500										8.61	3.36	1.53	.77	.42	.25	.09	.04	.02
589	3000											4.82	2.19	1.11	.61	.36	.14	.06	.03
686	3500											6.54	2.97	1.50	.83	.48	.19	.09	.04
786	4000												3.91	1.98	1 08	.63	.25	.11	.05
884	4500												4.94	2.51	1.36	.79	.32	.15	.07
984	5000												6 19	3.10	1 69	.99	.39	.18	.09
At 80 Pounds Gauge																			
7.74	50	14.31	3.96	1.53	.33	.10	.03	.01											
11.3	75		8.46	3.26	.71	.21	.08	.03	.01										
15.2	100		15.31	5 92	1.28	.39	.14	.06	.03	.02	.01								
19.4	125			9.64	2.09	.64	.24	.11	.05	.03	.01								
23.2	150			13.79	2.99	.91	.34	.15	.07	.04	.02	.01							
27.2	175				4.09	1.25	.47	.21	.10	.06	.03	.01							
31.0	200				5.34	1.63	.61	.27	.13	.07	.04	.01							
38.7	250				8.32	2.54	.96	.43	.21	.12	.07	.02	.01						
46.5	300				12.01	3.67	1.38	.62	.30	.17	.09	.03	.01						
54.2	350					4.99	1.88	.84	.41	.23	.13	.05	.02	.01					
62.0	400					6.53	2.45	1.11	.54	.30	.17	.06	.03	.01					
69.7	450					8.25	3.13	1.40	.69	.38	.22	.08	.04	.01					
77.4	500					10.81	3.83	1.73	.85	.47	.27	.10	.05	.02	.01				
92.9	600						5.61	2.46	1.22	.68	.39	.15	.06	.03	.02	.01			
108.2	700						7.46	3.37	1.66	.92	.53	.20	.09	.04	.02	.01			
124.0	800						9.86	4.42	2.18	1.19	.69	.27	.12	.06	.03	.02	.01		
139.5	900							5.61	2.77	1.54	.88	.34	.15	.08	.04	.02	.01		
152	1000							6.64	3.29	1.82	1.04	.40	.18	.09	.05	.02	.01		
232	1500							15.41	7.62	4.24	2.43	.95	.43	.22	.12	.06	.02	.01	
310	2000								13.62	7.58	4.32	1.69	.77	.39	.21	.12	.04	.02	.01
387	2500									11.79	6.88	2.64	1.19	.60	.33	.19	.07	.03	.01
465	3000										9.72	3.79	1.73	.87	.48	.28	.11	.05	.02
542	3500										13.25	5.27	2.35	1.19	.65	.37	.15	.06	.03
620	4000											6.78	3.07	1.55	.85	.49	.19	.09	.04
697	4500											8 54	3 89	1.97	1 08	.66	.25	.11	.05
774	5000											10.55	4.79	2.46	1.33	.77	.30	.14	.07

For longer or shorter pipes the friction loss is proportional to the length, *i.e.*, for 500 feet ½ of the above; for 4,000 feet four times the above, etc.

Delivery in Cu. Ft. of Compressed Air per Min.	Equiv. Delivery in Cu. Ft. of Free Air per Min.	Size of Pipe, Inches: 1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	16
		Loss of Pressure, in Pounds, by Friction in 1000 Feet Lengths of Pipe																	
At 100 Pounds Gauge																			
6.41	50	11.89	3.29	1.28	.27	.08	.03	.01											
9.61	75		7.42	2.87	.62	.19	.07	.03	.01										
12 81	100		13.20	5.11	1.15	.34	.12	.05	.02	.01									
15.81	125			7.75	1.68	.52	.19	.08	.04	.02	.01								
19.22	150			11.42	2.48	.76	.29	.13	.06	.03	.02	.01							
22.39	175				3.36	1.03	.39	.17	.09	.04	03	.01							
25.62	200				4.43	1.36	.51	.23	.12	.06	.04	.02	.01						
31.62	250				6.72	2.06	.77	.35	.17	.09	.05	.02	.01						
38.44	300				9.95	3.04	1.14	.51	.25	.14	.08	.03	.01						
44.78	350				13.41	4.11	1.54	.69	.34	.19	.11	.04	.02	.01					
51.24	400					5.40	2.06	.92	.45	.25	.15	.05	.03	.01					
57.65	450					6.85	2.57	1.16	.57	.32	.18	.07	.03	.02	.01				
63.24	500					8.21	3.08	1.39	.68	.38	.22	.08	.04	.02	.01				
76.88	600					12.21	4.58	2.14	1.03	.57	.33	.12	.05	.03	.02	.01			
89.56	700						6.19	2.79	1.38	.77	.44	.17	.07	.04	.02	.01			
102.5	800						8.13	3.67	1.81	1.00	.57	.22	.10	.05	.03	.02	.01		
115.3	900						10.23	4.64	2.29	1.27	.76	.28	.13	.06	.04	.02	.01		
126.5	1000						12.39	5.00	2.76	1.23	.88	.34	.16	.08	.04	.03	.01		
192.2	1500							12.81	6.68	3.51	2.03	.78	.36	.18	.09	.05	.02	.01	
256.2	2000								11.35	6.61	3.62	1.41	.67	.33	.18	.10	.04	.02	.01
316.2	2500									9.50	5.51	2.14	.97	.49	.27	.16	.06	.03	.01
384.4	3000									14.04	8.11	3.16	1.44	.76	.39	.23	.19	.04	.02
447.8	3500										10.95	4.26	1.93	.98	.53	.31	.12	.05	.03
512.4	4000										14.48	5.59	2.55	1.30	.72	.41	.16	.07	.04
576.5	4500											7.04	3.22	1.84	.89	.52	.21	.09	.05
632.4	5000											8.51	3.88	1.93	1.07	.63	.25	.11	.08
At 125 Pounds Gauge																			
5.26	50	9.88	2.70	1.05	.23	.07	.03	.01											
7.89	75	22.20	6.07	2.37	.51	.16	.06	.03	.01										
10.51	100	39.50	10.82	4.22	.91	.28	.10	.05	.02	.01									
13.15	125		16.88	6.58	1.42	.43	.16	.07	.04	.02	.01								
15.79	150		24.33	9.47	2.04	.63	.23	.11	.05	.03	.02	.01							
18.41	175		33.05	12.90	2.78	.85	.32	.14	.07	.04	.02	.01							
21.05	200			16.84	3.63	1.11	.42	.19	.09	.05	.03	.01							
26.30	250			26.30	5.68	1.73	.65	.29	.15	.08	.05	.02	.01						
31.58	300			37.90	8.18	2.51	.94	.42	.21	.12	.07	.03	.01						
36,81	350				11.08	3.39	1.27	.58	.28	.16	.09	.04	.02	.01					
42.10	400				14.51	4.44	1.67	.75	.37	.21	.12	.05	.02	.01					
47.30	450				18.38	5.61	2.11	.95	.47	.26	.15	.06	.03	.01	.01				
52.60	500				22.68	6.95	2.61	1.18	.58	.32	.18	.07	.03	.02	.01				
63.20	600					10.00	3.76	1.69	.84	.46	.27	.10	.05	.02	.01	.01			
73.70	700					13.60	5.11	2.31	1.14	.63	.36	.14	.06	.03	.02	.01			
84.20	800					17.80	6.68	3.01	1.49	.83	.47	.18	.08	.04	.02	.01	.01		
94.70	900						8.45	3.81	1.88	1.04	.60	.23	.11	.05	.03	.02	.01		
105.1	1000						10.42	4.71	2.32	1.29	.74	.29	.13	.07	.04	.02	.01		
157.9	1500						23.48	10.59	5.23	2.90	1.65	.64	.29	.15	.08	.05	.02	.01	
210.5	2000							18.81	9.30	5.15	2.94	1.15	.52	.26	.15	.08	.03	.02	.01
263 0	2500							29.40	14.52	8.05	4.60	1.80	.82	.41	.23	.13	.05	.02	.01
315.8	3000								20.90	11.59	6.63	2.59	1.18	.60	.33	.19	.07	.03	.02
368.1	3500								28.51	15.78	9.01	3.53	1.61	.81	.45	.26	.10	.04	.02
422.0	4000									20.61	11.80	4.61	2.19	1.06	.58	.34	.13	.06	.03
473 0	4500									26.10	14.90	5.83	2.65	1.34	.73	.43	.17	.08	.04
526.0	5000									32.20	18.45	7.20	3.27	1.65	.90	.53	.21	.10	.05

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standard built moisture traps which may be purchased from various manufacturers.

Moisture traps should be placed at low points in the air line.

Reheating Compressed Air

Reheating the air is a simple process and serves the double purpose of eliminating annoyance due to freezing as a result of the moisture which the air may contain, and at the same time increasing the air volume.

When the warm compressed air leaves the compressor and passes through the receiver and the transmission lines to the point of use, the temperature of the air is considerably reduced. This represents a loss in volume. If the air were used immediately adjacent to the point of production, this loss would not be experienced, but the latter condition seldom, if ever, prevails.

Reheating the compressed air, at the point where it is to be utilized, a temperature of 250 deg. F., which is quite usual, results in expanding it in volume from 30 to 35 per cent, at a cost of about ⅛ of the heat units required to produce the same volume by compression. This approximation is based on the use of effective form of reheater, and the use of the air in the cylinder of the engine immediately after passing from the reheater.

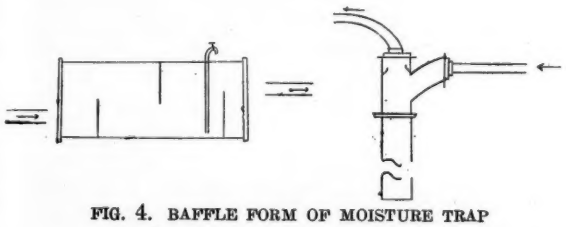

FIG. 4. BAFFLE FORM OF MOISTURE TRAP
FIG. 5. RECEIVER TRAP FOR MOISTURE

In using the reheater, therefore, it should be placed as closely as possible to the point of use of the air, to prevent loss from radiation of heat after the reheating process.

The reheater may consist of a cast-iron chamber containing a coil of pipe exposed to a fire, or a current of hot gas or steam. Figure 6 shows a cross-section view of one standard type of air reheater, which requires no explanation.

Power Utilized in Compressing

Two tables shown in Fig. 7 show the horsepower required for compressing air in single stage and two stage compressors at various gage pressures. All the conditions of the calculations given are explained in the headings.

Mechanical Efficiency

Mechanical efficiency of an air compressor is represented by the ratio of the indicated air horsepower to the indicated steam horsepower in the case of a steam-driven machine, and to the brake horsepower in the case of a power-driven machine.

The difference between the two represents excess power required to overcome frictional resistance of the machine, which is largely dependent upon the design, the workmanship and the supervision received when in operation.

Mechanical efficiency of the well designed air compressor ranges between 90 and 95 per cent.

Let us assume a power-driven air compressor of the following size and displacement and determine its mechanical efficiency:

Diameter low-pressure cylinder, 16 in.; diameter high-pressure cylinder, 10 in.; stroke, 14 in.; r.p.m., 165 in.; discharge pressure, gage, 100 lb.; piston displacement, cu. ft. per min., 534 lb.

From brake test it was found that the brake horsepower is 93.

The air indicator cards give an indicated horsepower of 83.6.

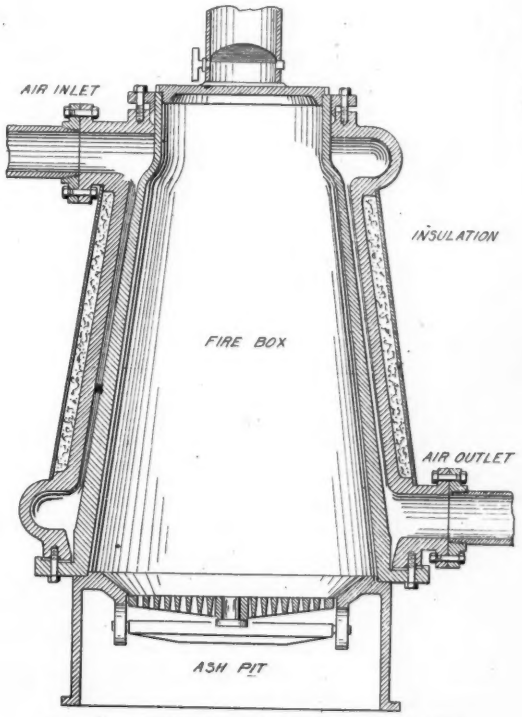

FIG. 6. A STANDARD FORM OF AIR REHEATER

Therefore $83.6 \div 93 = 90$ per cent mechanical efficiency.

As stated above, loss in mechanical efficiency is determined by frictional resistance. This is a factor which can be largely controlled by the supervising engineer. In general it is desirable that attention be given to the lubrication system to insure that all rubbing parts are sufficiently lubricated at the following points: Bearings, crossheads, stuffing boxes and cylinders, as they constitute the chief source of loss due to friction.

Bearings should be drawn up only enough to insure smooth operation, without undue pressure. Instructions covering the packing of stuffing boxes have already been given. Crosshead guides must be free enough to permit the crosshead to operate without undue pressure, and yet work without knocking.

for 4,000 feet four times the above, etc.

FIG. 7. HORSEPOWER TO COMPRESS 1 CU. FT. OF FREE AIR A MIN. FROM ATMOSPHERE (1417 LB.) TO GAGE PRESSURES SHOWN. INITIAL TEMPERATURE 60 DEG. F. NO JACKET COOLING. A FOR SINGLE STAGE. B FOR 2 STAGE

Gauge Pressure Pounds	Absolute Pressure Pounds	Number of Atmospheres	Isothermal Compression	Adiabatic Compression														Percentage of Saving over One-stage Comp.
			Mean Effective Pressure	H. P.	Mean Effective Pressure Theoretical	Mean Eff. Pressure plus 15% Friction	H. P. Theoretical	H. P. plus 15% Friction	Gauge Pressure Pounds	Absolute Pressure Pounds	No. of Atmospheres	Correct Ratio of Cylinder Volumes	Intercooler Gauge Pressure	Mean Eff. Pressure	H. P.	Mean Eff. Pressure Theoretical	Mean Eff. Pressure plus 15% Friction	H. P. Theoretical	H. P. plus 15% Friction	
5	19.7	1.34	4.13	0.018	4.46	5.12	0.019	0.022	.50	64.7	4.40	2.10	16.2	21.80	0.095	24.30	27.90	0.106	0.123	10.9
10	24.7	1.68	7.57	0.033	8.21	9.44	0.036	0.041	60	74.7	5.08	2.25	18.4	23.90	0.104	27.20	31.30	0.118	0.136	11.3
15	29.7	2.02	11.02	0.048	11.46	13.17	0.050	0.057	70	84.7	5.76	2.40	20.6	25.70	0.112	29.31	33.71	0.128	0.147	12.3
20	34.7	2.36	12.62	0.055	14.30	16.44	0.062	0.071	80	94.7	6.44	2.54	22.7	27.52	0.120	31.44	36.15	0.137	0.158	13.8
25	39.7	2.70	14.68	0.064	16.94	19.47	0.074	0.085	90	104.7	7.12	2.67	24.5	28.33	0.126	33.37	38.36	0.145	0.167	14.2
30	44.7	3.04	16.30	0.071	19.32	22.21	0.084	0.096	100	114.7	7.80	2.79	26.3	30.30	0.132	35.80	41.20	0.156	0.180	15.0
35	49.7	3.38	17.90	0.078	21.50	24.72	0.094	0.108	110	124.7	8.48	2.91	28.1	31.40	0.137	37.20	42.80	0.161	0.185	15.2
40	54.7	3.72	19.28	0.084	23.53	27.05	0.103	0.118	120	134.7	9.16	3.03	29.8	32.66	0.142	38.44	44.20	0.168	0.193	15.6
45	59.7	4.06	20.65	0.090	25.40	29.21	0.111	0.127	130	144.7	9.84	3.14	31.5	33.75	0.147	39.86	45.83	0.174	0.200	16.3
50	64.7	4.40	21.80	0.095	27.23	31.31	0.119	0.136	140	154.7	10.52	3.24	32.9	34.67	0.151	41.28	47.47	0.180	0.207	16.7
55	69.7	4.74	23.90	0.100	28.59	33.23	0.126	0.145	150	164.7	11.20	3.35	34.5	35.59	0.155	42.60	48.99	0.186	0.214	18.4
60	74.7	5.08	24.80	0.108	30.10	35.91	0.133	0.153	160	174.7	11.88	3.45	36.1	36.30	0.158	43.82	50.39	0.191	0.219	19.0
65	79.7	5.42	25.70	0.112	33.57	38.59	0.146	0.168	170	184.7	12.56	3.54	37.3	37.20	0.162	44.93	51.66	0.196	0.225	19.3
70	84.7	5.76	26.62	0.116	35.00	40.25	0.153	0.175	180	194.7	13.24	3.63	38.8	38.10	0.166	46.05	52.95	0.201	0.231	19.5
75	89.7	6.10	27.52	0.120	36.36	41.80	0.159	0.182	190	204.7	13.92	3.73	40.1	38.80	0.169	47.16	54.22	0.206	0.236	20.1
80	94.7	6.44	28.21	0.123	37.63	43.27	0.164	0.189	200	214.7	14.60	3.82	41.1	39.50	0.172	48.38	55.70	0.210	0.241	
85	99.7	6.78	28.93	0.126	38.89	44.71	0.169	0.195	210	224.7	15.28	3.91	42.8	40.10	0.174	49.38	56.70	0.216	0.247	
90	104.7	7.12	29.60	0.129	40.11	46.12	0.175	0.201	220	234.7	15.96	3.99	44.0	40.70	0.177	50.30	57.70	0.220	0.252	
95	109.7	7.46	30.42	0.132	41.28	47.46	0.180	0.207	230	244.7	16.64	4.08	45.3	41.30	0.180	51.30	59.10	0.224	0.257	
100	114.7	7.80	31.16	0.137	42.56	48.89	0.190	0.218	240	254.7	17.32	4.17	46.6	41.90	0.183	52.25	60.10	0.228	0.262	
110	124.7	8.48	32.60	0.142	45.09	51.83	0.199	0.229	250	264.7	18.00	4.24	47.6	42.70	0.186	52.84	60.76	0.230	0.264	
120	134.7	9.16	33.75	0.147	45.69	52.87	0.208	0.239	260	274.7	18.68	4.32	48.8	43.00	0.188	53.85	62.05	0.235	0.270	
130	144.7	9.84	34.67	0.151	49.64	57.08	0.216	0.249	280	284.7	19.36	4.40	50.0	43.50	0.190	54.60	62.90	0.238	0.274	
140	154.7	10.52	35.59	0.155	51.47	59.18	0.224	0.258	290	294.7	20.04	4.48	51.1	44.00	0.192	55.50	63.85	0.242	0.278	
150	164.7	11.20	36.30	0.158	53.70	61.80	0.234	0.269	300	314.7	20.72	4.55	52.2	45.30	0.197	56.70	64.15	0.246	0.282	
160	174.7	11.88	37.20	0.162	55.60	64.00	0.242	0.278	350	364.7	24.80	4.98	58.5	47.30	0.206	60.15	69.16	0.262	0.301	
170	184.7	12.56	38.10	0.166	57.20	65.80	0.249	0.286	400	414.7	28.20	5.31	63.3	49.20	0.214	63.19	72.65	0.276	0.317	
180	194.7	13.24	38.80	0.169	58.80	67.70	0.256	0.294	450	464.7	31.60	5.61	67.8	51.20	0.223	65.93	75.81	0.287	0.329	
200	214.7	14.60	39.50	0.172	60.40	69.50	0.263	0.303	500	514.7	35.01	5.91	72.1	52.70	0.229	68.46	78.72	0.298	0.342	

Air Compressor Efficiencies

DIFFERENT measurements necessary for calculation of efficiencies are given on the diagrams shown in Figs. 8 and 9.

Following are stated the ratios of these measurements resulting in the respective efficiencies and performance measurements of an electrically driven air compressor and a steam driven air compressor.

For electrically-driven air compressor,

B.H.P.
——— = Motor Efficiency
E.H.P.

I.H.P. Air Cylinder
——— = Mech. Eff.
B.H.P. on Shaft

For steam-driven compressor,

I.H.P. Air Cylinders
——— = Mechanical Efficiency.
I.H.P. Steam Cylinders

Free air act'ly delv'd
——— = Actual Volumetric Efficiency
Piston Displacement

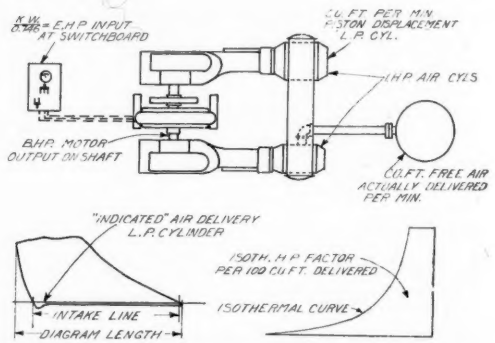
K.W. OR H.P. = E.H.P. INPUT AT SWITCHBOARD
CU. FT. PER MIN. PISTON DISPLACEMENT L.P. CYL.
I.H.P. AIR CYLS
B.H.P. MOTOR OUTPUT ON SHAFT
CU. FT. FREE AIR ACTUALLY DELIVERED PER MIN.
"INDICATED" AIR DELIVERY L.P. CYLINDER
ISOTH. H.P. FACTOR PER 100 CU. FT. DELIVERED
INTAKE LINE
ISOTHERMAL CURVE
DIAGRAM LENGTH

FIG. 8. PERFORMANCE MEASUREMENTS FOR ELECTRICALLY-DRIVEN AIR COMPRESSOR

Length L.P. Intake Line
——— = Indicated Volumetric Eff.
Length Indicator Diagram

Actual Volumetric Efficiency
——— = Slippage Efficiency
Indicated Volumetric Efficiency

I.H.P. air cylinder × 100
——— = I.H.P. per 100 cu. ft. delivered per min.
Cu. Ft. actual delivered per min.

E.H.P. input at switchboard × 100
——— = E.H.P. per 100 cu. ft. delivered per min.
Cu. Ft. actual delivered per min.

Isothermal Factor
——— = Isothermal Compression Efficiency
I.H.P. per 100 cu. ft. delivered per min.

Isothermal Factor
——— = Overall Isothermal Compression Eff.
E.H.P. per 100 cu. ft. delivered per min.

Lbs. Steam per min. × 1000
——— = Lb. steam per 1000 cu. ft. air delivered
Cu. ft. free air delivered per min.

All compression efficiencies should be subject to cor-

rection to perfect intercooling if specified amount of water at specified temperature is not available.

Compound vs. Simple Air Compressors

The theory of compound or stage compression is very readily understood. It is generally conceded that compound or stage compression should be employed, (a) with small capacity machines when pressures range over 125 lb., (b) with larger capacity machines when pressures range above 70 lb. The heat of compression increases with the pressure, therefore, the higher the pressure, the more difficult it is to reduce the temperature of the air to a point where efficient compression conditions and proper air cylinder lubrication obtain. For instance, in compressing air to 100 lb. terminal gage pressure in a single cylinder, the final temperature would be about 485 deg. F. Some of this heat would be absorbed by the cylinder walls and water jackets, but the final temperature would remain too high to insure efficient compression conditions and proper lubrication.

In Fig. 10 is shown a theoretical combined indicator

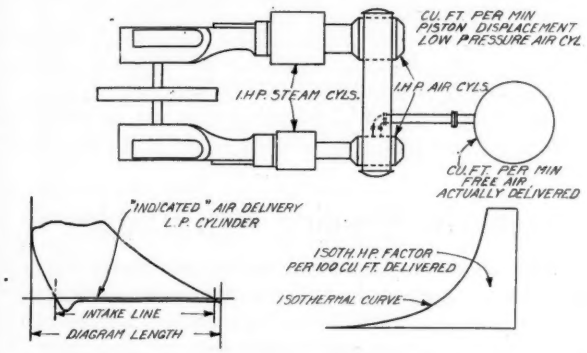
CU. FT. PER MIN PISTON DISPLACEMENT LOW PRESSURE AIR CYL.
I.H.P. STEAM CYLS.
I.H.P. AIR CYLS.
CU. FT. PER MIN FREE AIR ACTUALLY DELIVERED
"INDICATED" AIR DELIVERY L.P. CYLINDER
ISOTH. H.P. FACTOR PER 100 CU. FT. DELIVERED
ISOTHERMAL CURVE
INTAKE LINE
DIAGRAM LENGTH

FIG. 9. PERFORMANCE MEASUREMENTS FOR STEAM-DRIVEN AIR COMPRESSOR

diagram from a two-stage compressor. It is assumed that the compression follows the adiabatic curve in both cylinders, with perfect intercooling between.

In Fig. 10, AB = volume of low pressure cylinder; CD = volume of high pressure cylinder; both are drawn to the same scale. ACF = pressure to some designated scale; BEK (adiabatic curve) = relation between pressure and volume for any piston position, with no intercooling and no radiation through the cylinder walls.

In this curve the product $PV^{1.41}$ is constant (1.41 is the ratio between the specific heat of air at constant pressure and constant volume), where P = pressure and V = volume. BDH (isothermal curve) = relation between pressure and volume provided the temperature of the air under compression could be kept constant so that the product PV would also be constant.

Air taken into the low-pressure cylinder at zero gage pressure, is compressed along the adiabatic curve BE until at E it attains a pressure of 26.3 lb., equal to that of the intercooler, which allows the discharge valves to open and the air to pass into the cooler. In the intercooler, the volume of a definite weight of the air is reduced from CE to CD, so that the volume entering the high-pressure cylinder is less to the extent of DE, which is the same scale as CD and AB. This means that the given weight of air represented by the volume CE at a gage pressure of 26.3 lb., when cooled to the same temperature at which it was originally taken into the low-pressure air cylinder will be reduced in volume to CD provided the pressure of 26.3 lb. remains constant.

Air taken into the high-pressure air cylinder at 26.3 lb. is compressed along the adiabatic curve DG. At G the high-pressure discharge valves open and the air is discharged into the receiver at the desired gage pressure of 100 lb.

The shaded portion represents the power saving effected by the intercooler.

The saving due to the reduction in temperature between stages is not the only advantage of stage compression. The maximum temperature in each cylinder is reduced to a point where the heat can be more thoroughly drawn off by the water jackets surrounding the cylinder walls; also, the lower temperature in the cylinder is an insurance of good lubrication of piston, valves, etc.

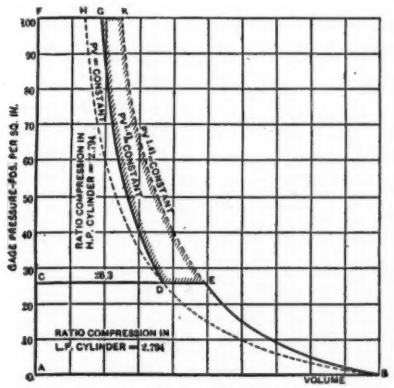

FIG. 10. THEORETICAL COMBINED INDICATOR DIAGRAM FOR 2-STAGE COMPRESSOR

As already indicated, the theoretical function of the intercooler in stage compression is to reduce the temperature of the air after it leaves the low-pressure cylinder to the same point at which it was first taken into that cylinder, before it enters the next stage cylinder.

Under practical working conditions, however, it will be found that the majority of coolers fail to accomplish this result and a reduction to within 5 to 10 deg. of the original temperature is considered a good accomplishment.

The intercooler consists of a shell containing a nest of tubes arranged in series or groups through which water flows successively, entering the bottom row and finding an outlet through the top row. Surrounding the water tubes is a series of baffle plates, which direct the flow of the air so as to split it up into thin films and insure intimate contact with the cold water tube surfaces. By reducing the spacing of these baffles as the high-pressure cylinder is approached the most efficient velocity of air in transit is attained.

When the air comes in contact with the cold tube surfaces, condensation of the moisture occurs, and it drips and flows down into the bottom of the shell. In

order to drain this condensed water the intercooler is usually placed at an incline, so that the water will flow into the pocket surrounding the water separator where a drain cock is provided for the occasional removal of the water.

Temperature changes in the intercooler cause the tubes to expand and contract, and in order to provide for this it is customary for the best of tubes to be fixed at one end only, while the other end, including the water box, is left free to move with the tube plate.

Air Compressor Regulation

As for any other piece of machinery, the load factor of an air compressor is the average amount of work actually done, divided by the maximum capacity for

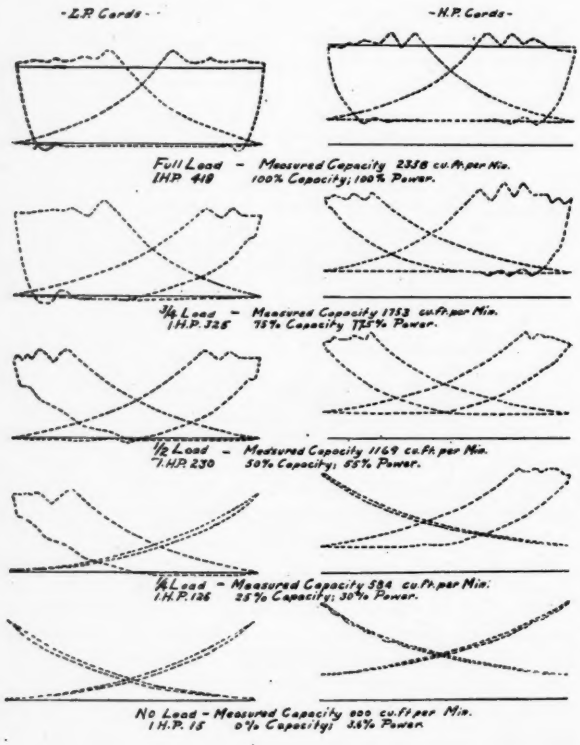
-L.P. Cards-
-H.P. Cards-
Full Load - Measured Capacity 2338 cu.ft.per Min.
I.H.P. 418 100% Capacity; 100% Power.
¾ Load - Measured Capacity 1753 cu.ft.per Min.
I.H.P. 325 75% Capacity 77.5% Power.
½ Load - Measured Capacity 1169 cu.ft.per Min.
I.H.P. 230 50% Capacity; 55% Power.
¼ Load - Measured Capacity 584 cu.ft.per Min.
I.H.P. 126 25% Capacity; 30% Power.
No Load - Measured Capacity 000 cu.ft.per Min.
I.H.P. 15 0% Capacity; 3.6% Power.

FIG. 11. AIR COMPRESSOR DIAGRAMS AT VARIOUS LOADINGS

work. The result represents the commercial efficiency of the machine, or in other words, the overall cost of production.

It is quite usual to install an air compressor to take care of the maximum demand and depend upon some automatic means of regulation or unloading to vary the output of the machine to fluctuating needs. This varies the horsepower input with the load and effects a saving in cost of operation.

It is further quite usual where the load will vary materially, especially in electrically-driven plants of large capacity, to install a number of small units and vary the number operating, keeping them at full load, to comply with changes of load, to assist in maintaining the highest overall economy. In such cases, regulation depends upon automatic starting and stopping.

The reduction in horsepower input with power-driven air compressors controlled by some form of partial unloading device is not in direct ratio to the reduction in output. For instance, in Fig. 11 is shown a series of cards taken from a 28 by 17½ by 21-in. electrically-driven two-stage air compressor, operating at 185 r.p.m. and equipped with a four-step clearance regulator. It will be noted that at three-quarter load capacity, the horsepower input is 77.5 per cent; at half load, the horsepower input is 55 per cent; at quarter load, it is 30 per cent, and at no load 3.6 per cent. The difference is accounted for by the fact that the mechanical efficiency drops off rapidly with a decrease in load; for example, in this particular case at full load the efficiency is 94 per cent, three-quarter load 92 per cent, half load 87.5 per cent, and at quarter load 80 per cent.

Air compressor control devices are usually set by the manufacturer to meet the conditions imposed by the particular installation, and as the design of these devices varies with the different manufacturers, instructions of only a very general nature can be given here.

If the device fails to function properly, it is probably due to an obstruction in the way of dirt or other foreign substance, and not to a need for readjustment. It should be thoroughly cleaned and care taken in reassembling to adhere to the adjustment existing prior to the cleaning.

Collective Dealing with Employes

IN an article in Automobile Industries, Harry Tipper calls attention to the fact that from time immemorial the general method of paying workers has been on a time basis, and that this system will probably prevail as the most acceptable for a long time to come.

In Great Britain, not until 1824 was the right of private contract between employer and worker recognized. Wages and conditions of employment were controlled by legislation, which provided against combinations of workers for increasing wages or changing conditions.

Since 1860 the need of collective bargaining has been recognized, resulting in standards of wage and standards of performance, the latter based on the ability or preference of the majority rather than the recognition of the individual or his skill.

Based on these facts, Mr. Tipper presents the argument that unless an employer is willing to accept the principle of collective dealing with his employes, he will have to face the alternative of bargaining collectively with the general body of workers in whom he is not interested and who can have no interest in the success of his concern. Mr. Tipper advocates the creation of a workers' committee in the factory to be recognized in settling questions of wages, hours, working conditions and other matters affecting the workers. This is along the lines of the plans adopted by the British Reconstruction Commission, and he feels that it is the one course which can succeed in introducing some form of payment that shall give recognition to the individual.

Dealing in America's Greatest Assets

PROFESSIONAL DIVISION OF THE UNITED STATES EMPLOYMENT SERVICE ASSISTS EXPERIENCED TECHNICAL MEN OF THE ARMY AND NAVY TO FIND THEIR PLACES IN WORK OF RECONSTRUCTION

AMERICA'S greatest assets, the brain power and energy of her thoroughly trained young men, are the commodities in which the Professional Division of the United States Employment Service is dealing. Officers and men of the army and navy released from active service are being registered with the Division, and placed in touch with those employers who can best make use of their services.

The Professional Division deals only with those men who are well equipped by education and experience in their particular lines of work. The record of each man is carefully investigated before registration is permitted. Many university graduates in mechanical, electrical and civil engineering, and in chemistry, and other technical men with several years of practical experience, have already been registered. These men, who willingly severed their business relations more than a year ago to give their services to their country, are returning to civil life to find changed conditions. Although the industry of the country has great need of their services, neither men nor employers are able, without assistance, to discover each other immediately. To avoid delay in the readjustment processes, not only the labor of the country but also the highly trained directors of industry are being mobilized with the assistance of the government. The aim is that each man shall fit in that part of our business organization where he can do his best work.

The task of dealing with thoroughly trained men who in many instances can command high salaried positions requires the assistance of those technical organizations which have heretofore placed university graduates and experienced men with employers. The professional Division is seeking to co-operate with all such societies by referring properly qualified men to them, or by obtaining from them data on positions available.

The engineering field appears to present the largest problems of the Professional Division. Thus far, nearly one-half of all the applicants have been qualified for work in various forms of the engineering profession. The temporary lull in general construction work has in part closed one field which, it is believed, will be more available by the time the overseas forces begin demobilization on a large scale.

The Professional Division of the United States Employment Service has its New York offices at 16 East Forty-second Street. Its registrations of experienced men are increasing. Employers seeking such men are asked to inform the Professional Division of the precise nature of the positions which they have available. Only those men who are well qualified to fill such positions are referred to the employer.

THE PORTLAND CEMENT ASSOCIATION, 111 West Washington St., Chicago, announces the appointment of Wm. M. Kinney as general manager.

Tax Anticipation Certificates

ASIDE from maintaining public order and the defense of life and property, the most vital function of Government is the provision of money, or an acceptable medium of payment and exchange.

The financial needs of the Government, in war and after that for the restoration of the peace footing, are very large. Heavy taxation is necessary and the problem of meeting the Federal taxes without affecting the equilibrium of money and credits vital to the welfare of the nation. To meet the condition of raising six billion dollars of taxes in 1919 the Government has decided on two policies: (1) Installment payment of the taxes on incomes and profits—the first installment falling due March 15. (2) The issuance of Tax Anticipation Certificates bearing 4½ per cent interest and maturing at the time the taxes must be paid.

In effect the Tax Anticipation plan enables all who are liable for taxes to the Treasury to rally their cash means in the interval between now and March 15, 1919, and to invest that cash in a perfectly safe Government security, at a fair rate of interest, and in a form which will be accepted by the Treasury in tax payments March 15, 1919. Under this plan it will be possible to avoid any disturbance of the money market and credit conditions at the time of the enormous tax payments next spring.

The matter is thus summed up by R. G. Elliott:

"War taxes are in the nature of a division of profits, with the Government in the position of a preferred stockholder who must be paid before there can be a distribution of profits to the owners; therefore, provision should be made to cover this tax liability, the the money withdrawn monthly or quarterly and invested in liquid form during the year in which the profit is earned.

"These Treasury certificates furnish a convenient form of income-producing reserve and by investing in them during the taxable year to the extent of the tax liability, makes each year stand on its own bottom and will tend to defer the issue of bonds and permit an earlier reduction of war taxes for Governmental expenditure, bringing us closer to home in our financing by reducing expansion, thus creating a sounder and more stable credit condition."

GOVERNMENT control of the railroads for a period of 5 yr. is now advocated by Director General McAdoo. He argues that the period of 21 mo. after signing of a peace treaty is so short a time that the imminence of return to private control will break down the morale of the organizations, and will not permit of financing the necessary improvements and extensions and carrying them to completion under government management. He favors either immediate return to the owners, which is considered impossible without legislation to change present regulations in regard to operation, or legislation which will extend the government operation period to a term of 5 yr. to allow effective test of this system under peace conditions and time for discussion and decision as to the best plan for the future.

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Did the Engineer Cause the War?

Many philosophical gentlemen have called attention to the greater material comfort that has been made available to humanity by the inventions of engineers and the application of power in place of muscular effort. Not a few have contended that this has resulted in a tendency of the race toward materialism and the seeking for the ownership and enjoyment of material things rather than for the higher ideals of action and character. Certainly, the mechanical facilities for making war, heavy guns, highpower rifles, explosives, airplanes, tanks, submarines, which the engineer has contributed are responsible for the vastly greater loss of life in the recent war than in any preceding conflict.

Engineering of London calls attention, however, to the work of George Stephenson and his successors as the impulse to ambitious rulers to extend their influence and control, and cites the ability given to Russia to bind Siberia to her, the consequent friction with Japan, and the temptation to Germany to attempt the control of the Balkan rail route to the East as the main incentives in bringing on the two great modern wars.

So long as foot or horse was the only means of travel, it was difficult for a ruler to keep in touch with far-flung domains and to maintain his power over them. Colonies that could be reached by ships were the goal, therefore, of the autocrat who sought to enlarge his empire. With perfection of railroads, lands away from the sea became available, and the joining of great inland states into empires was possible if the ruler could establish his might over them.

Creation of great inland cities became common, with the massing of the people, resulting in their becoming accustomed to following leaders and taking orders instead of thinking and acting as individuals. Under autocratic governments this developed to the point where the people thought more of their bodily comfort than of their rights and freedom, hence were easily controlled and even deluded. This came as the consequence of large use of machinery and power, and to that extent the engineer has been responsible for war.

But he and his works must also be credited with winning the war for democracy and liberty. If his inventions made possible the seducing of great masses or of nations to the immoral purposes of autocratic rulers, they also made possible the welding together of free nations to resist the onslaught of those rulers. By his devices the thought and purpose of the Allies was consolidation; by them the cordon was established which shut in the central powers from outside aid; by them the menace of Zeppelin, gas and submarine was overcome, and forces gathered from all over the earth sufficient to throw back the armies of autocracy in defeat: anti-aircraft guns, gas masks, nets, depth bombs, listening devices to detect the approach of a submarine, heavy and heavier guns, methods for rapid production of shells and building of ships. For these the engineer must have credit, so that, if his work made possible the effectiveness of the German war machine, it also provided the means to overcome it.

The engineer has been welcomed and lauded; he has been given first place in helping to win the war. Is

he to be ignored, now that he has accomplished the task? That will depend much on the members of the profession. If they insist on recognition of their importance in time of peace, they will be recognized. If they are content to sink back to an interest in developing machinery and processes for the benefit of mankind, and leave executive and governmental activity to lawyers, capitalists and politicians, they will soon find themselves in the background as before, doing the will of the more aggressive professions.

Carrying On in Fuel Conservation

The threatened shortage of fuel this winter, and the memory of last winter's shutdowns are sufficient proof that we should hold fast what has been gained in the way of fuel economies and continue our efforts to find other ways to augment them, both as a matter of present reduction of expense and to postpone future rise in price which must come as our sources of cheap fuel are diminished so that more remote sources and those more difficult to work or of lower grade furnish our chief supply.

One encouraging fact as to the possibility of making savings, but discouraging in view of past effort to get such waste eliminated is that the largest economies have been effected by preventing waste of heat and power after they have been delivered from the boiler and engine rooms. This includes waste of live and exhaust steam in heating of buildings and industrial processes; motors underloaded or running idle, heavy friction losses, poor belts, lights burning idle, uncovered pipes, leaks, machinery operating under capacity. These matters have been remedied largely by supervision of a Fuel and Power Committee in each establishment to look for the wastes and arouse a public spirit among employes decrying such waste.

Careful attention to burning of fuel and best use of steam in prime movers has, of course, been another source of saving. But President Main of the A. S. M. E. has called attention to larger problems that must be studied in the future. These include study of the proper place and manner of production of power for all purposes, which involves the question of advisability of large steam plants at or near coal mines, or on tide water where coal and oil may be easily brought by vessels, and electric transmission to manufacturing centers; development of hydro-electric power wherever it can be done economically; use of waste heat from blast furnaces and coke ovens; re-development of old and inefficient water power plants; storage of flood waters so as to conserve their power; and interconnection of transmission and distribution systems to better the load factor.

These developments may require governmental action in one or more of three ways; to assist in development where the investment involved is too great for private enterprise; to modify regulations and restrictions so as to permit a reasonable return on capital invested, while safeguarding the interests of the government and the public; to encourage or even compel co-operative action by companies whose systems lie parallel or adjacent in order to secure the best service to the public and greatest economy of operation.

Such action should be taken with regard not only to reduction in the cost of power, but to the conservation of natural resources and securing reliability of service to consumers, in order to secure the greatest good for the community or country.

This, by no means, suggests the diminishing of power generation by individual or by block plants. But such a policy would insure that each such plant was justified, was doing the best work possible, and that where power is supplied from a central source it is done because that is the cheapest and best method; not because of plausible arguments, personal prejudice or outside influence. The millenium is not at hand or in sight, but the war has taught us the superior importance of the rights and welfare of the whole people as compared to the selfish interests of one or a few individuals

Authorities Contributing to This Issue

In order that readers may appreciate the authoritative character of the articles appearing in the Jan. 1 issue, we append a list of the authors and the positions which they hold:

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Geo. S. Blankenhorn, Engineer, Wilson-Snyder Mfg. Co.

O. C. Callow, Mechanical Engineer of the Mark Mfg. Co.

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Carl L. Svensen, Director of the Ohio Technical School.

R. E. Turner, Associate Editor, Power Plant Engineering.

F. F. Vater, President of the Power Plant Specialty Co.

H. A. Wilcox, Consulting Engineer.

J. Wilson, Engineer Steam Turbine Dept., Allis-Chalmers Mfg. Co.

News Notes

L. E. TOWNSLEY, Mgr. Foreign Trade Department, Paul R. Ruben & Co., 16 California St., San Francisco, Cal., leaves the last of January for China, Japan, Philippines, Dutch East Indies, Straits Settlements and Australasia, to establish resident agents at principal ports. Correspondence is invited with American manufacturers desiring to increase their foreign trade.

SPECIFICATIONS for the preliminary work on the Los Angeles, Calif., new power plant No. 2, in the San Franciscquita canyon, were taken up recently at the meeting of the public service commission. The work will cost between $50,000 and $100,000. The work on the power plant will be rushed to completion with all speed. It will, when finished, add 28,000 hp. of electrical energy to the city's power system.

THE ENTIRE classification yards of the New Haven road are to be electrified, and while the work is under way the temporary supports for the electric system from the west out northward are to be replaced by permanent supports of steel. In connection with the extension of electricity a new power house may be required, though it is likely that if the power house at Zoar Bridge is finished within two years the road may buy its power from that source and so save the expense that a new power house would incur.

Catalog Notes

CATALOG NO. 244 of the Jeffrey Manufacturing Co., which has just been published, demonstrates the advantages of using Jeffrey Standardized elevators and contains other data of interest, 40 pages being devoted to details of elevators selected out of numerous styles used in the handling of a wide range of materials in practically every industry of the country. A page is given to each Standard elevator which is illustrated, both in perspective and in line drawing, giving dimensions. There is also an illustration showing the chain and bucket used in that type of elevator, and at the bottom of each sheet is given a list of specifications.

GENERAL ELECTRIC CO.'S office calendar for 1919 consists of six large leaves carrying one month on each page. It is of convenient form, having below the current month, the preceding and succeeding months for reference. It is being sent out from the general office at Schenectady, N. Y., on request.

LINK-BELT CO., 39th St. and Stewart Ave., Chicago, is sending out its new illustrated 52-page Book No. 353, Economical Handling of Coal and Ashes and Reserve Coal Storage.

BULLETIN 107, relating to the Vater water softening system, has been received from Power Plant Specialty Co., 1306 Monadnock Block, Chicago.

FROM GREENFIELD Tap and Die Corporation, Greenfield, Mass., we have just received No 40 catalog of small tools. This catalog lists that company's line of taps, dies, screw plates and reamers.

The other lines manufactured by the company are taken care of in a separate catalog, as follows: Pipe Tools, No. 38; Machine Tools, Nos. 39 and 41; Gages, in course of preparation.

A pocket edition of No. 40 catalog is now ready and will be supplied to jobbers and dealers for general distribution.

Trade Notes

CHAIN BELT CO. of Milwaukee, Wis., has established a branch office at 1018 Corn Exchange Bank Bldg., Chicago, with Herbert C. Follinger in charge.

W. F. KECKEISEN, formerly advertising manager of the International Filter Co., and previously associated with the Federal Sign Co., has joined the staff of Russell T. Gray, advertising engineer, Chicago.

CHAMPION AND RYAN, 299 Broadway, New York, have recently arranged to act as eastern sales representatives for Ajax Electric Specialty Co., of St. Louis, Mo., covering the territory embracing Boston, New York, Philadelphia and export trade.

PRATT & CADY CO., Hartford, Conn., has purchased the entire plant and business of I. B. Davis & Sons, of Hartford, manufacturers of feed-water heaters and hot-water heaters. The company will continue the manufacture of the Davis products.

L. E. SCHUMACHER, who for the past 8 yr. has been chief inspector of the Westinghouse Electric and Manufacturing Co. at East Pittsburgh, Pa., has been promoted to works manager of the Krantz Manufacturing Co., of Brooklyn, N. Y., the latest subsidiary of the former company.

YARNALL-WARING CO. has acquired a three-acre tract at Mermaid Lane and Devon St., Chestnut Hill, Philadelphia, adjacent to Mermaid Station, Chestnut Hill division, Philadelphia and Reading Railway. The tract contains a three-story stone mansion which is being converted into general offices, and a one-story modern machine shop has been erected in which manufacturing is now being carried on.

AT A SPECIAL MEETING of the Board of Directors of Pratt & Cady Co., changes in official personnel of the company were made as follows:

Edwin L. King was elected treasurer to succeed the late Bishop White. He will also continue to act as secretary. Additional offices have been created and filled by the election of O. Lamson Beach as assistant secretary, and Albert E. Oldroyd as assistant treasurer. Mr. Beach has been assistant sales manager for the last year and a half, and Mr. Oldroyd has been chief accountant since 1914.

In addition to the above-mentioned officers, W. B. Lashar is president, and B. I. Ashman is vice-president and general manager.